AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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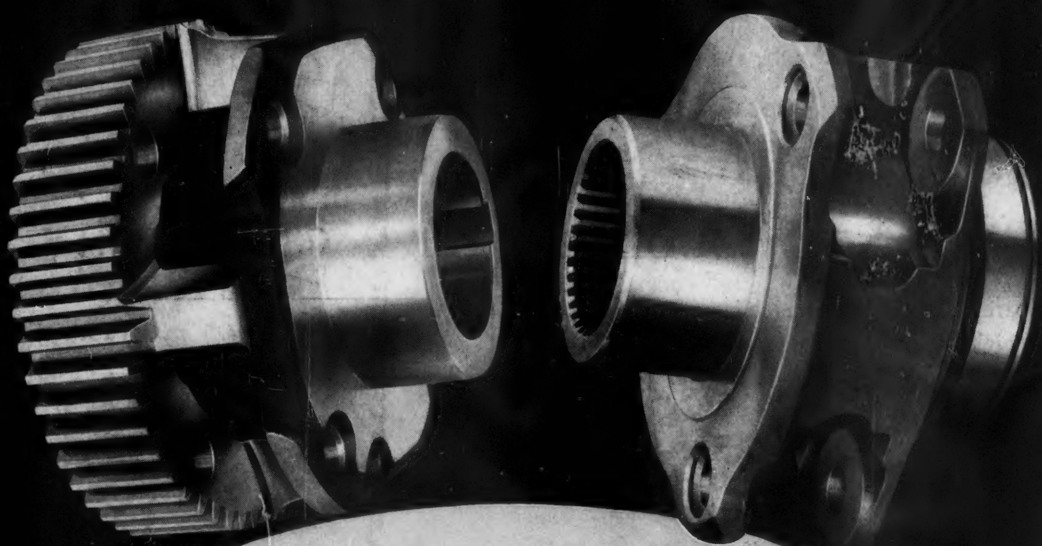

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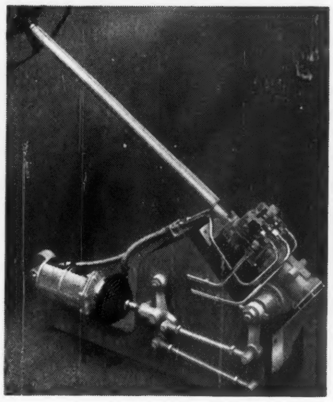

A CLAYTON DEWANDRE PNEUMATIC POWER-ASSISTED STEERING SYSTEM IN WHICH THE CONTROL VALVE ASSEMBLY IS INCORPORATED IN THE STEERING COLUMN

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F. C. Sheffield

DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Chassisless ?

LARGELY as a result of impending increases in insurance rates, the controversy over the relative merits of chassisless, or unitary, construction and chassis with separate frames has again come to the fore. However, it is doubtful if the current high cost of repairs to accidental damage can be entirely, or even largely attributed to the adoption of unitary construction, rather is it the outcome of modern stylistic trends.

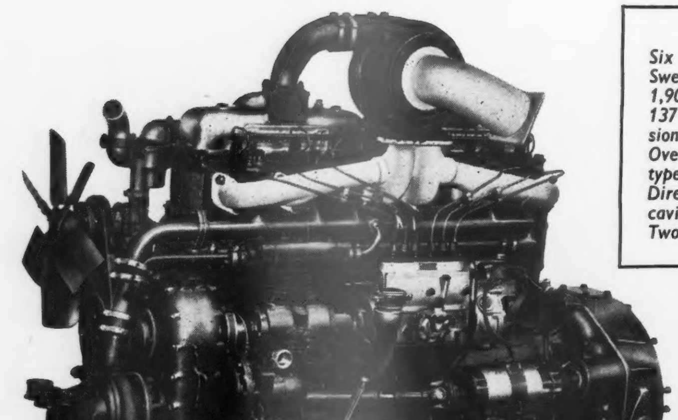
Before the War, damage caused by minor accidents was generally confined to wings, running boards and radiator grilles. In most vehicles all these components were generally easy to repair or replace and, since they were almost invariably relatively small, the cost of replacement was not high. On the other hand, as is well known, modern cars have wing panels that are large and, in some instances, welded in position; they have no running boards to fend off glancing blows commonly experienced in dense traffic in our cities; and radiator grilles are securely fixed to complex pressings that form the bonnet surround. With full-width bodywork, accidental damage frequently extends over a large proportion of the length of the vehcle. Nevertheless, bodywork in this style has obviously come to stay since it enables the best possible use to be made of the available space.

Few manufacturers now dispute the fact that unitary construction, because it is lighter and costs less than the traditional layout based on a separate frame, is the most suitable for the relatively small European cars produced in large quantities. In the design of a separate frame, the provision of adequate strength presents no problems, but stiffness in torsion and bending can only be obtained by the employment of heavy components. In short, chassisless vehicles, if properly designed and constructed, are both stronger and stiffer than vehicles of the same weight with a separate frame. For European markets, the increases in fuel consumption and in prime cost, which inevitably accompany increases in weight, are, of course, unacceptable.

Motorists are often heard to exclaim that they feel safer with a good heavy frame underneath them, but this sentiment does not seem to influence them to the extent that many are prepared to pay the higher prices of most of the cars with separate frames. In any case, it is questionable whether they are really safer in that type of vehicle. A heavy frame is extremely rigid, so far as its resistance to horizontal impacts is concerned, whereas the thinner sections of a chassisless vehicle tend to buckle and cushion the shock of a collision. The extent of this buckling is rarely great enough for the main structure to be pushed against the occupants and injure them. However, the steering column is sometimes forced back against the driver; when a remedy for this defect can be found, a major advance will have been made towards the reduction of serious injuries in motor accidents. In most instances, if the structural damage sustained by a chassisless car in an accident is serious enough to warrant replacement of the major components, the frame, had there been one, would also have been distorted, so the cost of repair might have been much the same. Corrosion, which used to be a serious problem, can now be overcome by the application of modern protective treatments.

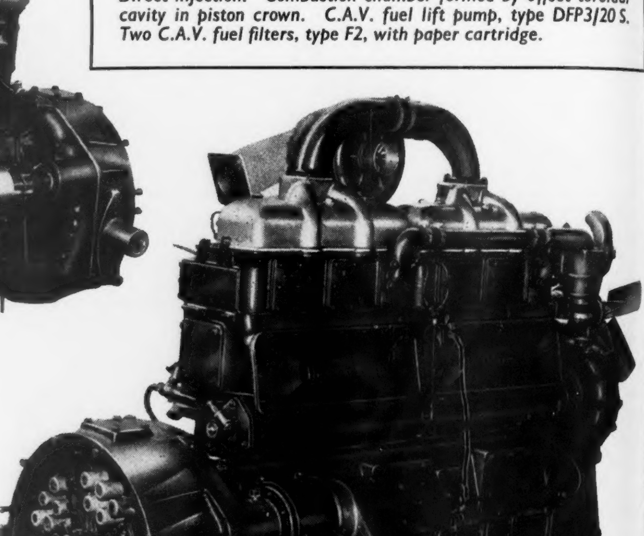
Manufacturers tend to recall wistfully the days when they used to be able to effect major styling changes without any retooling for the chassis. Nevertheless, this can still be done, as has been demonstrated by the Continental coachbuilders who have made a wide variety of special bodies on small, quantity-produced chassis of unitary construction. The reason why British coachbuilders have not followed suit is simply that they provide for an entirely different type of market. Despite this, there are good reasons why the large manufacturers should design their cars in such a way that their basic structures of unitary construction are suitable for conversion by coachbuilders: the main advantage to the chassis manufacturer would be that styling changes could be effected more readily.

It is often asked why many of the American manufacturers have not adopted unitary construction more widely. There are at least three reasons for this. One is that when a separate frame is employed, different models, such as four- and two-door convertibles, hard-tops, family saloons and estate cars can be produced readily on the same basic structure. The second reason is that with vehicles having such a long wheelbase, it is difficult to design a chassisless structure with adequate stiffness for the four-door convertible style of body. Thirdly, styling changes do not involve any major alterations to self-contained chassis. The problem of obtaining a low roofline, while still maintaining adequate headroom above the frame, is solved in many instances by the incorporation of wells in the floor panels. This is not difficult in the large American vehicles, because there is so much more space between the side members of the frame than there is in the European cars. Americans accept the weight penalty, since slight increases in fuel consumption are unimportant, because of the relatively low cost of petrol in their country.



SPECIFICATION

Six cylinders. Bore and stroke $4\frac{3}{4}$ in and $6\frac{1}{2}$ in respectively. Swept volume 691 in³ (11·33 litres). Maximum b.h.p. 200 at 1,900 r.p.m. (governed). Maximum b.m.e.p. and torque 137 lb/in² and 625 lb-ft respectively at 1,000 r.p.m. Compression ratio 16:1. Forged, seven-bearing crankshaft, fully balanced. Overhead valves, operated by push rods. C.A.V. injection pump, type NNR 6F 90, and C.A.V. injector nozzles, type BDLL 150S. Direct injection. Combustion chamber formed by offset toroidal cavity in piston crown. C.A.V. fuel lift pump, type DFP3/20 S. Two C.A.V. fuel filters, type F2, with paper cartridge.



Most of the auxiliary units are mounted on one side of the engine, only the starter motor being on the other side. This is a good feature from the point of view of accessibility for the servicing of these units

Thornycroft 11·33 *Litre Oil Engine*

A Unit That Can be Supplied in Naturally Aspirated or Turbo-Charged Form

THE Thornycroft KRN6 oil engine and its turbo-charged counterpart, the KRN6/S, are 11·33 litre six-cylinder power units of substantially identical design. Indeed, they differ only in the matter of whether or not a supercharger is installed and in the form of the intake and exhaust manifolding dictated by this alternative equipment. Although the engine first went into production in its naturally aspirated form, its design was developed initially with the ultimate object of increasing its torque and output by at least 30 per cent by the application of turbo-charging. Accordingly the construction of the engine when it is considered as a naturally aspirated unit appears in some respects unusually robust and its bearing areas generous.

Primarily the two closely related engines, which are the largest produced by Transport Equipment (Thornycroft) Ltd., are employed in the Thornycroft Big Ben range of rigid six-wheeled chassis. This range extends from load-carriers having a gross weight rating of 31 tons up to tractors with gross train weight ratings of 60 tons when coupled to a semi-trailer. They have been evolved as heavy-duty transport vehicles with arduous conditions, met in undeveloped areas overseas, particularly in mind. There are four-wheel-driven and six-wheel-driven variants of the chassis, choices of several lengths of wheelbase, versions with either normal or forward driving controls and with either left- or right-hand steering. It is in the largest tractors of this comprehensive series that the KRN6/S engine is utilized; the KRN6 unit is installed in the smaller capacity machines.

In general terms the design of the basic power unit can be described as conforming with current British direct-injection oil engine practice. The KRN6/S engine is, however, exceptional in the method of supercharging it embodies. After considerable research and experimental work concerned with the question of supercharging the decision was taken to adopt the Eberspächer exhaust-driven turbo-charger, which is now to be manufactured under licence in this country by Simms Motor Units Ltd. One of the advantageous characteristics of a turbo-charger of this class is that the thermal and kinetic energy of the exhaust gases, which would otherwise be wasted, are utilized to drive it, and this obviates the need to introduce a mechanical drive, which absorbs energy otherwise available as useful power. A subsidiary advantage of the turbo-charger is considered to be that it damps out exhaust pulsations to some extent and thus tends to reduce exhaust noise.

Geratebau Eberspächer of Esslingen, the German manu-

facturers of this particular turbo-charger, have collaborated closely with Thornycroft to achieve the improvement in torque and output characteristics required by the engine designers. The application of the turbo-charger so developed, increased the maximum torque from 508 lb-ft to 625 lb-ft at 1,000 r.p.m., and the maximum output from 155 b.h.p. to 200 b.h.p. at the governed engine speed of 1,900 r.p.m. At the same time, the increase in specific fuel consumption was not unacceptable to transport operators. This improvement in performance is accompanied by the addition of only 40 lb to the weight of the engine.

The bore and stroke dimensions are $4\frac{3}{4}$ in and $6\frac{1}{2}$ in respectively, giving a swept volume of 691 in^3 and a stroke : bore ratio of 1·37 : 1. A connecting rod : stroke ratio of 2·15 : 1 has been adopted, and at 1,900 r.p.m., the speed at which maximum b.h.p. is developed, the mean piston speed is 2,060 ft/min. The compression ratio is 16 : 1. At 1,000 r.p.m. the maximum b.m.e.p. of 137 lb/in^2 is developed. In terms of b.h.p/in^2 piston area the output is 11·3 and in terms of b.h.p/litre it is 17·65. Since the dry weight of the engine, less clutch, dynamo, starter and the air compressor usually fitted for vehicle braking, is 2,620 lb the output is 0·076 b.h.p/lb. A minimum specific fuel consumption of 0·365 lb/b.h.p-hr is obtained. The overall dimensions of the engine less air filter are:—height, $59\frac{11}{16}$ in; width, $32\frac{1}{8}$ in; and length, $64\frac{7}{16}$ in measured between the rear face of the flywheel housing and the front of the fan.

Separate castings are employed for the cylinder block and crankcase; this is because of the size of the engine and also to simplify machining operations. Advantage is taken of this feature to manufacture the engine in alternative forms. In the interests of accessibility for routine service attention, more particularly in those installations in which the drivers seat and controls are alongside the engine, auxiliaries such as the fuel pump, air compressor, dynamo and the lubricating oil filters are all mounted on one side of the crankcase.

To suit right-hand driving control layout, this grouping of the main auxiliaries is on the left of the engine, the starter motor being on the right; also, the cylinder block is assembled with the camshaft-tappet group, which it embodies, on the right. This arrangement can, however, be reversed, to meet the requirements of a left-hand driving control chassis layout. To enable this to be done the cylinder block is designed so that it can be mounted the other way round on the crankcase; the faces machined on one side of the crankcase to receive the mounting brackets for the auxiliaries are duplicated on the other.

The timing gear casing is a separate assembly and alternative design patterns are produced to match left- and right-handed engines. As a further variation to suit different installations, the engine can also be supplied with either of two designs of sump. The most usual pattern embodies a well at the rear, while in the alternative the well is at the front. In both cases the design is such that the lubrication system will continue to operate satisfactorily when the engine is inclined up to 25 deg from the horizontal, which it might be called upon to do when the vehicle traverses gradients of that order in extreme cross-country transport conditions.

Crankcase

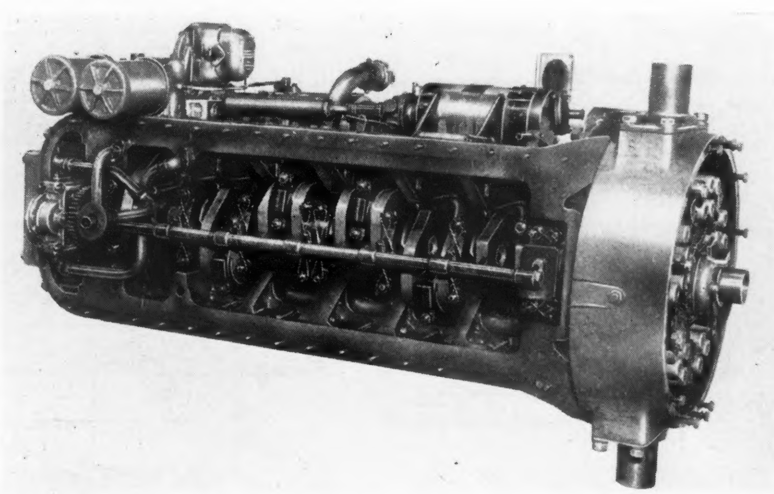
The crankcase is a chromium iron casting. It is 13 in deep and its skirt extends 5 in below the level of the crankshaft axis. A series of internal ribs, vertically disposed to correspond with and support the crankshaft bearing seats, contribute to the overall stiffness of the casing. The top decking is $8\frac{3}{4}$ in wide and $\frac{3}{4}$ in thick. Locally, the thickness of the decking is increased to $1\frac{7}{16}$ in by six internal bosses equally disposed below the deck on both sides. These bosses are drilled and reamed to receive $\frac{5}{8}$ in diameter fitted studs and the holes are counter-bored to provide recessed seats to accommodate $\frac{7}{8}$ in diameter collars formed on the studs. The studs are pulled down from below by slotted nuts. At their upper ends, the studs project $1\frac{1}{4}$ in above the face of the crankcase; they are supplemented by eight $\frac{5}{8}$ in diameter studs screwed into tapped holes in the decking on both sides, to provide the means for securing the base of the cylinder block rigidly in position.

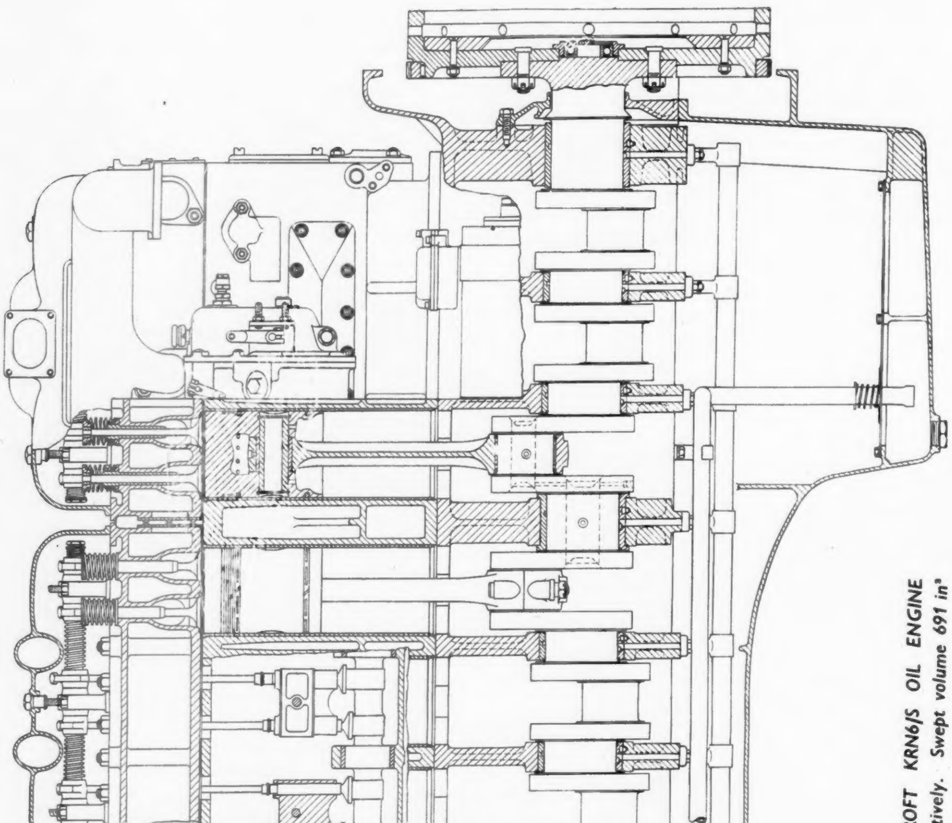
At the rear of the crankcase there is a flanged extension machined to afford a face against which the upper portion of the flywheel housing is secured, the lower portion of the housing is held against a corresponding face machined on a flanged extension at the rear of the LM4M sump casting. Location of the flywheel housing is effected by two $\frac{3}{4}$ in dowels on the crankcase flange. These dowels are diametrically disposed relative to the crankshaft axis. The housing is pulled up on its facing by nuts on ten $\frac{7}{16}$ in diameter studs.

Cylinder block

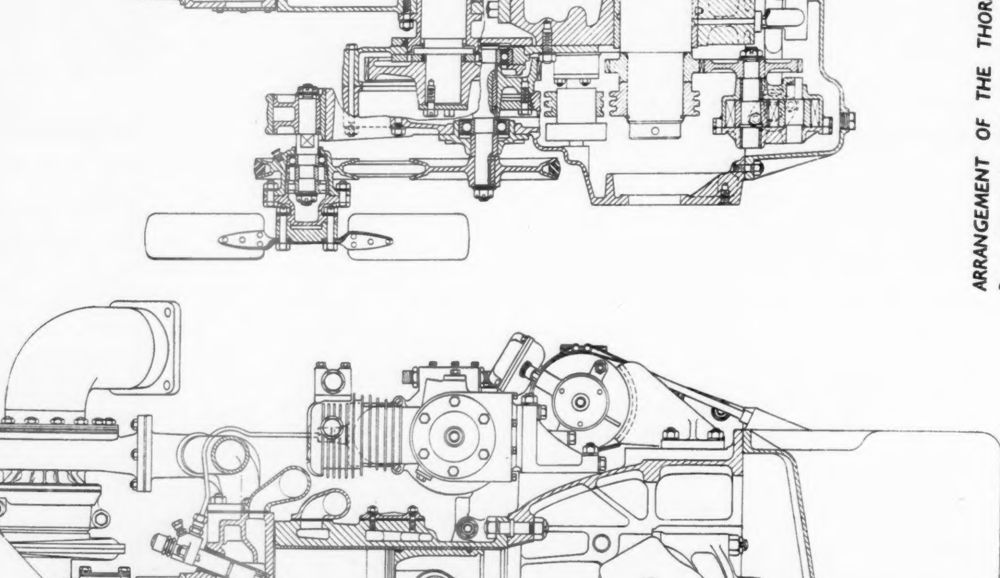
A 13 in deep chromium iron casting forms the cylinder block. The thickness of the top decking is $1\frac{1}{16}$ in and the flange through which the crankcase securing studs pass, at the base of the block, is $\frac{3}{4}$ in thick. The cylinder liner housings are integral with the block. Their wall thickness is $\frac{1}{4}$ in and there is a minimum water jacket space of $\frac{5}{8}$ in

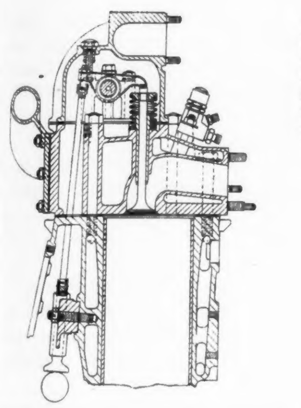
Adequate rigidity of the crankcase is ensured by the incorporation of substantial internal ribs. The crankshaft-driven oil pump is mounted at the forward end





ARRANGEMENT OF THE THORNYCROFT KRN6/S OIL ENGINE
Bore and stroke $4\frac{3}{4}$ in and $6\frac{1}{2}$ in respectively. Swept volume 691 in^3





The valves are carried in the conventional manner, vertically in line, but the layout is unusual in that the inlet port entry is at the top. The exhaust ports, which are shown sectioned in this illustration, discharge at the side of the cylinder head

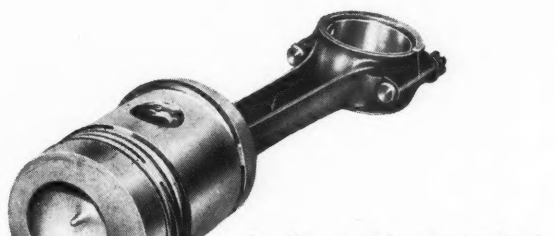
between them. Centrifugally cast chromium iron liners, 12⅞ in long, and having a wall thickness of ⅛ in, are pressed into the cylinders. The interference fit is 0·004-0·0055 in. After insertion the liner bores are hone finished to 5-10 micro-in. The width and thickness of the liner flanges, which seat in recesses at the cylinder head end of the bores, is 0·182 in by 0·125 in. One side of the cylinder block embodies a tunnel for the camshaft. This tunnel is combined with an open trough for the valve tappet block assemblies.

Crankshaft

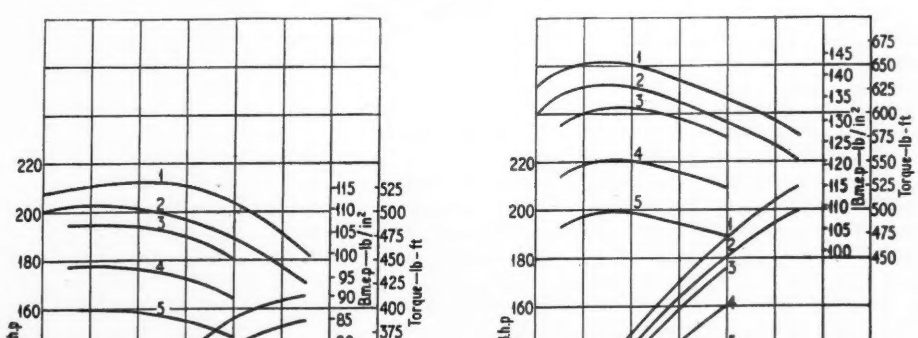
A seven bearing En 41T crankshaft is employed; its journals and crankpins are nitride hardened. Measured from the forward face of the front web to the back of the rear web, the length of the crankshaft is 36½ in. Each web is 1·062 in thick and 6¼ in wide. A 13¼ in diameter Holset viscous torsional vibration damper is attached by four 7/16 in diameter fitted bolts to the front web of the shaft. The journals are 3·9970-3·9962 in diameter and their effective bearing lengths are: front 2·75 in; centre 2·809 in; rear 3·375 in; and the remainder 1·375 in.

For lubrication purposes, each journal is bored 1⅞ in diameter longitudinally; both ends of the bores are chamfered to form seats for concave aluminium closure discs. These discs are drilled centrally and pulled on to their seatings by ⅜ in diameter tie bolts and slotted nuts, and thus seal the bore of the journal. The bolt heads and nuts, beneath which are aluminium washers, do not project beyond the faces of the crank webs. A similar arrangement is employed for the crankpins: their diameter is 3·125-3·1245 in and they are longitudinally bored to 1⅜ in diameter. In this case, however, the axis of the bore instead of corresponding to that of the crankpin is offset radially outwards by ⅛ in. Sealing is effected in the same way by cupped aluminium discs drawn against their seatings by 5/16 in tie bolts and slotted nuts, with aluminium washers beneath the nuts and bolt heads. Drilled radially in the crank webs are 3/16 in diameter ducts to communicate between the bores in the journals and those in the crank pins. A 3/16 in diameter radial hole in each crank pin, and two ¼ in diameter radial holes in each journal communicate with their respective bores.

Prefinished steel-backed main bearing shells, lined with copper-lead, are employed. Their diametrical running clearance is 0·0063-0·0045 in and their interference fit in the housings is 0·0015-0·0025 in. Thrust faces, formed on both ends of the centre bearing, locate the crankshaft axially. The main bearing caps are located and held down by ⅝ in diameter En 16T studs and slotted nuts locked by wire. Four such studs secure the caps of the front, centre and rear bearings, and two are employed for the caps of each intermediate bearing. Shell location is effected by a dowel in each bearing cap. Axial location of the upper half of the bearing is by a vertical peg at the side of the bearing seating. The peg registers in a semi-circular cut-out in the periphery of the upper half of the bearing shell. This arrangement



An offset, toroidal combustion chamber cavity is formed in the crown of the piston



1 *Peak output CS 102E-42*
2 *Net output CS 102E-42, automotive rating as installed in road vehicle*
3 *One-hour rating, B.S.649, intermittent output CS 102E-42*
4 *Twelve-hour rating, B.S.649*
5 *Continuous rating, B.S.649, horse power CS 102E-42*
a *Minimum speed for operation at continuous rated output*
b *Maximum speed for industrial operation*

Power curves for the turbocharged and naturally aspirated engines. The fuel consumption curve is given at the one-hour rating and all figures are corrected to a barometric pressure of 29·92 in of mercury and an ambient air intake temperature of 60 deg F

Piston Ring Data

	Top	Second	Third
Compression rings:			
Gap	0·022 in	0·017 in	0·017 in
Face width	0·124 in	0·124 in	0·124 in
Radial thickness	0·187 in	0·187 in	0·187 in
Depth of groove in piston	0·199 in	0·199 in	0·199 in
Side clearance	0·003 in	0·003 in	0·003 in
Oil control rings:			
Gap	0·017 in	0·017 in	
Face width	0·249 in	0·249 in	
Radial thickness	0·187 in	0·187 in	
Depth of groove in piston	0·199 in	0·199 in	
Side clearance	0·003 in	0·003 in	

Mean dimensions, between the tolerance limits, are given

has been adopted so that, once the associated bearing cap is detached, the upper half of each bearing can be rotated in its seating round the journal, and thus can be removed without dismantling the crankshaft assembly.

Behind the rear main journal, an oil thrower is machined round the crankshaft, and to the rear of the thrower there is a $\frac{7}{8}$ in thick by $9\frac{3}{4}$ in diameter flange. The flywheel, which is 3·687 in thick by 20 in diameter and weighs 113 lb, is secured to this flange by six $\frac{5}{8}$ in diameter fitted bolts and slotted nuts. Its moment of inertia is 52·8 lb-ft^2. The periphery at the forward face of the flywheel is shouldered to $18\frac{1}{2}$ in diameter to receive a starter ring-gear having 159 teeth. This gear is of En 18R and is shrunk on and located by a peg.

Keyed to the 3·125 in diameter front extension of the crankshaft are an En 16T gear pinion, for driving the oil pump, and a timing chain sprocket of the same material. Both are an interference fit on the shaft and are retained by a Seeger circlip.

Connecting rods and pistons

The connecting rods are I-section stampings of En 18R and have a centre-to-centre length of 14 in. Their cross-sectional dimensions are: width over the flanges $1\frac{5}{16}$ in; depth $1\frac{3}{4}$ in; and the minimum web thickness $\frac{3}{16}$ in. Each rod weighs 10 lb 9 oz complete with bearing caps, shell and bolts. Thin steel-backed, pre-finished copper-lead bearing shells are fitted in the big ends, which are divided at 90 deg to the axis of the rod. The big ends are narrow enough to be withdrawn through the cylinder bores. Each cap is located by the fit of its two $\frac{9}{16}$ in diameter En 16T holding down bolts, the nuts of which are locked by split pins. The length of the bearings is 1·875 in and the diametral clearance between them and the crankpins is 0·003-0·004 in. Positive location is effected by lugs, on the bearing shells, registering in corresponding slots in the rod and cap. A 1·875 in long phosphor bronze bush is employed in the small end. Its interference fit in the eye of the rod is 0·0028-0·007 in. The diametral clearance between the bush and the pin is 0·0003-0·0007 in. An En 36V gudgeon pin, case hardened to 700 V.P.N., is employed; its inside and outside diameters are 0·750 in and 1·625 in respectively. The gudgeon pin bearing length in each piston boss is 1·062 in, and axial location of the pin is effected by a Seeger circlip at both ends.

Wellworthy tin-plated Lo-Ex aluminium alloy pistons of the solid skirt type are employed; each weighs 7 lb. A toroidal cavity having a prominent central pip is formed in the crown. It is offset $\frac{5}{16}$ in from the centre of the piston to correspond with the offset of the tip of the injector nozzle relative to the cylinder bore. Each piston carries three square-faced compression rings, the upper of which is chromium plated. In addition there are two slotted oil control rings, one immediately below the lowest compression ring and the other near the base of the skirt.

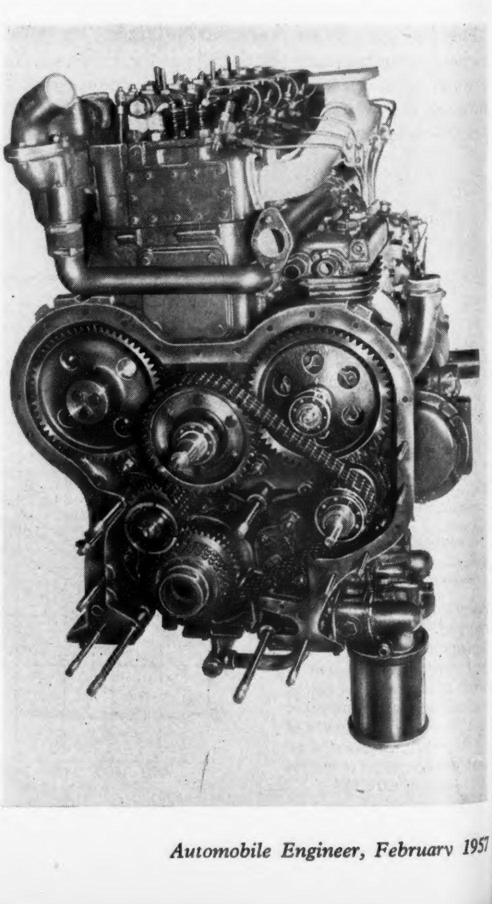
Timing drive and camshaft

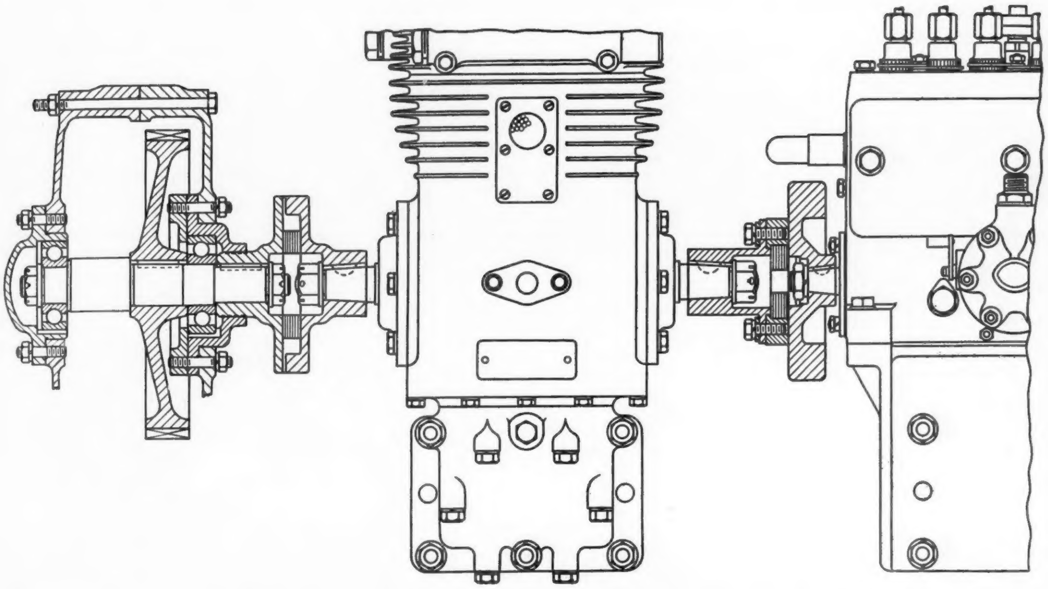
As has been mentioned earlier, alternative timing gear casings are made to suit right- and left-handed engines. The casing is not simply an extension to the crankcase, but is a separate iron casting pulled up to the front face of the crankcase and the sump by nine $\frac{1}{2}$ in diameter studs and slotted nuts. It has a cast iron cover to close its front end. Location of the rear half of the timing case is effected by two dowels, and there is a third location in the form of a spigot provided by a forward extension of the front camshaft bearing in the cylinder block.

An unusual combination, of a chain and helical gears, is employed for the timing drive. A Renolds $\frac{1}{2}$ in pitch triple-row roller chain transmits the drive from the crankshaft sprocket to an idler sprocket and the dynamo drive sprocket. The chain is automatically tensioned by a Renolds spring-operated eccentric type unit.

Bolted to the rear of the idler sprocket is a helical gear pinion which, on one side, meshes with the camshaft gear and, on the other, with the gear that drives the compressor and the fuel injection pump in tandem. The helical gear pinions are of En 8Q and have a helix angle of 27 deg 23 min 7 sec. All the chain sprockets are of En 16T. Both the shaft on which the idler sprocket and gear assembly is mounted and the dynamo drive shaft extend forward through

A combination of a triple-row roller chain and helical gear wheels is employed in the timing drive assembly. Bolted to the rear of the larger of the two driven sprockets is a helical gear pinion, which meshes on one side with the camshaft gear and on the other with a gear that drives the compressor and fuel injection pump, these two components being connected in tandem





C.A.V. flexible couplings connect the shafts of the compressor and injection pump, which are driven in tandem by a helical gear at the front

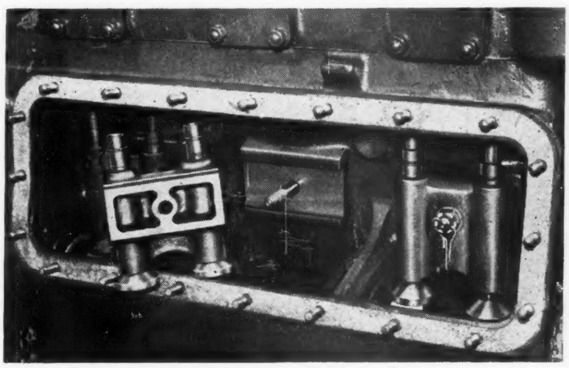
the timing gear casing, in which they are supported in ball bearings. They drive the fan belt pulley and the water pump respectively.

The half-speed wheel is keyed to the camshaft and retained by a plate secured to the end of the shaft by three $\frac{3}{8}$ in diameter set bolts, the heads of which are wired together. A $\frac{3}{8}$ in thick phosphor bronze thrust ring, through which the forward end of the camshaft extends, registers between the back of the half-speed wheel and the front face of a $3\frac{7}{16}$ in diameter collar machined forward of the front journal bearing of the camshaft. This ring is secured to the timing case by $\frac{3}{8}$ in diameter bolts. Between the cams, the diameter of the En 32 camshaft is $1\frac{9}{16}$ in. A $\frac{1}{2}$ in diameter oil duct is drilled axially through the shaft. There are seven, 2·496-2·495 in diameter journals. The effective length of the front one is $2\frac{9}{16}$ in and of the centre one $2\frac{3}{8}$ in; the length of the remaining journals is $1\frac{1}{8}$ in. A $\frac{3}{8}$ in wide, semi-circular oil groove is machined in the periphery of the front journal and is connected to the central duct in the camshaft by a $\frac{3}{8}$ in diameter radial hole. Radial oil holes, $\frac{3}{16}$ in diameter are drilled in the other journals.

An unusual feature is the employment of aluminium alloy camshaft bearings of LM 14 LP. Their diametral clearance is 0·004-0·006 in. The front bearing embodies a collar by means of which it is located against a face machined on the cylinder block. Three $\frac{7}{16}$ in diameter studs secure it. The outside diameter of the bearing is increased from 3 in to 4 in at its forward end to form a spigot to help to locate the timing gear case. Each of the remainder of the camshaft bearings is located by a dowel formed on the extremity of a $\frac{1}{2}$ in diameter set bolt screwed through the base of the camshaft tunnel.

The cams have an effective lift of 0·375 in and their nominal periods are: inlet 125 deg, exhaust 132 deg. When the tappet is on the flank of the cam, its maximum positive acceleration is approximately 1,807 ft/sec^2 and the maximum negative acceleration occurring at the nose of the cam is 922 ft/sec^2.

Mushroom-headed tappets of En 8Q, with Stellite faces, are employed. They are carried in pairs in cast iron tappet guide blocks, as shown in an accompanying illustration. The inner faces of the blocks are machined to fit in $1\frac{3}{4}$ in



This illustration shows one of the tappet blocks detached and turned round, to show the method by which it is secured to the chest

wide grooves in bosses in the tappet chest. Each tappet block is secured by a nut on a $\frac{7}{16}$ in stud.

The tappet guide diameter is 0·625 in and the diametral clearance between the tappets and their guides is 0·0025-0·0040 in. Inset in the top of the stem of each tappet is an En 24Z cup, which receives the spherical end of the CDS6 tubular push rod. The effective length of the push rods is 10·624 in, and their diameter is 0·375 in. A cup of En 24Z is fitted at the upper end of each rod to form a seat for the spherical end of the case hardened En 32 tappet adjusting screws. Two cover plates, each secured by 20 nuts and plain washers to studs in the side of the cylinder block, enclose the tappet chest.

Stamped En 8Q rockers, each with a Stellited pad at the valve end, are employed. Bronze alloy bushes, 1 in long, are pressed into the bores of the rockers, and the diametral clearance between the bushes and shaft is 0·0005-0·002 in. The En 32 rocker shafts, one on each cylinder head, are case hardened and have outside and inside diameters of 0·8745 in and $\frac{1}{4}$ in respectively. Their ends are sealed by En 3B screwed plugs.

Each rocker shaft is carried by three aluminium alloy

Valve Data

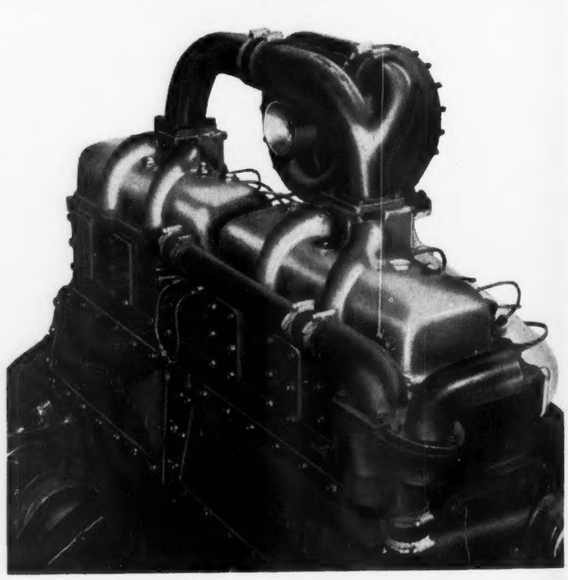
	Inlet	Exhaust
Material	En 51	En 52
Head diameter	1·875 in	1·750 in
Stem diameter	0·4335–0·4327 in	
Diametral clearance in guide	0·004–0·0056 in	
Seat angle	45 deg	
Valve seat material	Cast iron	
Spring material	En 49C	
Spring rate;		
inner	27·1 lb/in	
outer	41·4 lb/in	
Spring length, free;		
inner	2·45 in	
outer	3·10 in	
Spring length, installed at maximum load;		
inner	1·660 in	
outer	1·910 in	
Spring natural frequency;		
inner	13,500 c/min	
outer	11,080 c/min	
Number of coils;		
inner	6	
outer	6	
Coil diameter:		
inner	1·090 in, mean	
outer	1·356 in, mean	
Wire gauge;		
inner	0·110 in	
outer	0·144 in	
Valve lift	0·510 in	
Rocker ratio	1·37:1	
Valve guide material	Phosphor bronze	
Valve guide length	3·250 in	3·375 in
Valve guide;		
inside diameter	0·4375–0·4383 in	
outside diameter	0·6890–0·6840 in	
Tappet clearance	0·010 in	0·012 in
Valve opens	8 deg B.T.D.C.	39 deg B.B.D.C.
Valve closes	40 deg A.B.D.C.	13 deg A.T.D.C.

pedestals, and is located axially by a ⅜ in diameter, dowel-ended set bolt, passing radially through the front pedestal, to register with a hole in the shaft. This bolt is locked by a tab washer. Two ⅜ in diameter studs and nuts hold down each pedestal; one stud of each pair extends upwards and is fitted with a dome nut to secure the rocker cover. The rockers are constrained against the pedestals in the usual manner by coil springs.

Valve gear

All the valves are disposed vertically and in line in the cylinder heads. The distance between the axes of adjacent inlet and exhaust valve stems is 2·562 in. Shrouded inlet valves are employed to impart swirl to the air as it enters the cylinder. Two springs are fitted to each valve and they are retained at the top by a shouldered washer and split tapered collets held by circlips. The bottom ends of the springs seat on the shoulders of collars fitted over the upper end of the valve guides. The guides are of phosphor bronze; those for the exhaust valves are 3⅜ in long while those for the inlet valves are 3¼ in long, the former projecting 1 in and the latter ⅞ in into their respective ports. Further

Arrangement of the inlet manifolding used in conjunction with the Eberspächer turbo-charger, showing the way in which the inlet ports are taken to the tops of the two cast aluminium covers of the rocker chambers

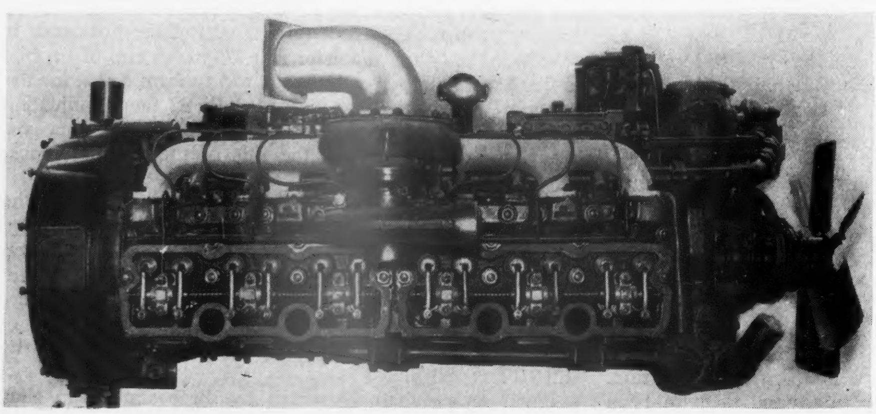


details of the valves, their springs and their guides are tabulated.

Cylinder heads

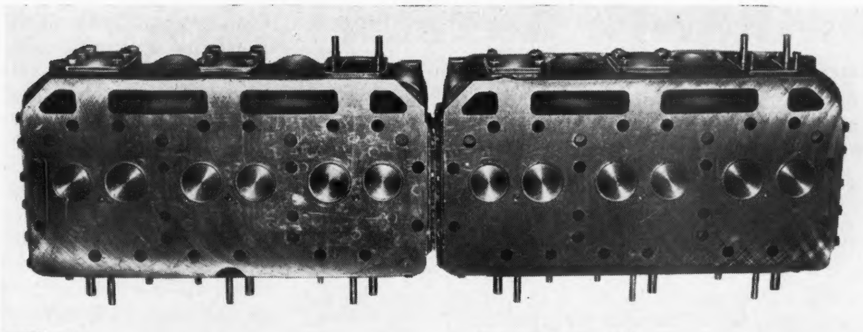
The two cast iron cylinder heads are interchangeable. Each is located and pulled down, by twenty $\frac{9}{16}$ in diameter En 16T waisted studs and nuts, on to a copper and asbestos gasket. Both the head castings have an overall depth of 5 in and an overall width of 10⅛ in. Composition washers are interposed between the heads and the rocker covers.

Separate exhaust and inlet ports are incorporated in the heads. The injectors, surrounded by copper sleeves, are set at an angle of 20 deg from the vertical, with the nozzle tip offset $\frac{5}{16}$ in from the centre of the cylinder bore. Although



Rocker covers removed to show the two entries to the three inlet ports in each cylinder head

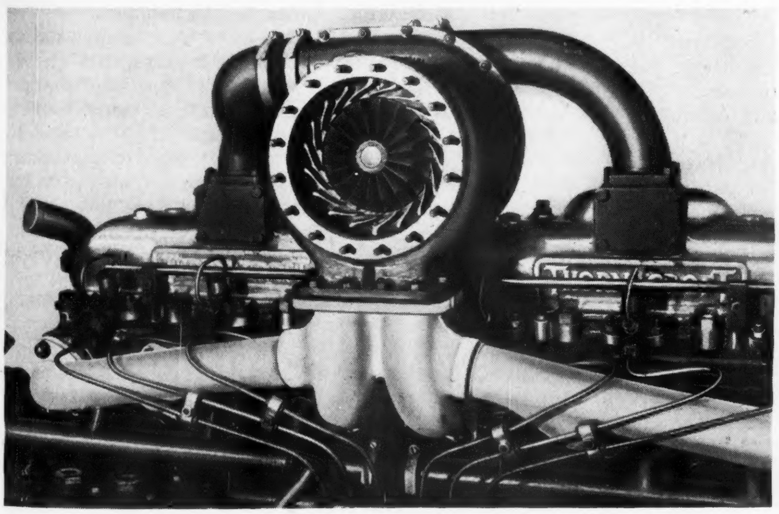
The cylinder heads are interchangeable and, as can be seen from this illustration, their bottom faces are machined flat



the exhaust port outlets are arranged on one side of the cylinder head, in the conventional manner, the entries to the inlet ports are on top.

There are two inlet entries on each head: one serves a single inlet port and the other serves the branches of the remaining two. Each inlet entry is fed by a corresponding air duct, which is embodied in the LM4M aluminium alloy casting of the rocker box and which stems from a single orifice in a boss on top of the rocker box. To this boss, the joint flange of the external inlet manifold is secured. Thus, the ducts incorporated in the rocker boxes can be regarded as forming part of the induction manifolding. The unfamiliar layout of the inlet ports and air ducting is not peculiar to the turbo-charged engine, but is also employed

Removal of the exhaust pipe elbow discloses the turbine rotor and nozzle ring of the Eberspächer turbo-charger. This unit is carried by the central portion of the exhaust manifold



in the naturally aspirated engine, although the external manifolds differ.

In the case of the turbo-charged engine, the LM4M inlet manifold extends in the form of an arch between its flanged joints on the rocker boxes. It has a pipe diameter of 2¾ in and is in two sections coupled centrally by a reinforced rubber hose and two clips. A branch pipe is embodied in the foremost section to provide a connection with the outlet from the compressor chamber of the supercharger.

The exhaust manifold is in three sections, with branch pipes of 1¾ in internal diameter, the central section being relatively short and incorporating branch pipes from numbers 3 and 4 cylinders. Bolted to the upper face of a horizontal outlet-flange is the corresponding flange at the base of the turbo-charger turbine casing. Thus the central section of the exhaust manifold provides the support for the turbo-charger unit. Branch pipes from numbers 1 and 2 cylinders are incorporated in one of the outer portions of the exhaust manifold, and those from numbers 5 and 6 cylinders in the other. These outer portions spigot into the centre section, so as to give freedom for expansion.

Turbo-charger

The overall dimensions of the complete Eberspächer turbo-charger unit are: height 11 1/16 in; width 10⅞ in; and length 10 in. In the Thornycroft engine installation, the axis of the unit is horizontal and set transversely above the cylinder heads, between the exhaust and inlet manifolds. The unit comprises a cast steel turbine casing at one end and a cast aluminium compressor casing at the other, joined by a central cylindrical member having a lubricating oil sump formed in its base. Within this assembly are a 4⅞ in diameter exhaust turbine rotor and a 5 5/16 in diameter centrifugal compressor secured to the extremities of a common shaft. This rotor assembly is 7 in long and weighs approximately 2·2 lb. It is free to rotate in ball bearings, which are lubricated by splash from the sump of the unit, which has a capacity of ¼ pt.

Within the turbine casing, and surrounding the periphery of the bladed turbine rotor, is a so-called nozzle ring comprising a static series of shaped blades to direct the exhaust gases tangentially inwards to the turbine blades. The exhaust gases enter through twin ports in the base and are directed round the inside of the chamber in to the blading of the nozzle ring; finally they leave the chamber in an axial direction through a 4 in diameter exhaust pipe elbow, whose flange is secured, by sixteen 8 mm diameter studs and nuts, to the outer face of the turbine casing.

The maximum rotational speed of the turbine and compressor in this application is about 35,000 r.p.m. At this speed, the compressor delivers air into the inlet manifold at 6·5 lb/in^2. The moment of inertia of the rotors of this unit is low, so the response of the turbine to changes in requirements is almost instantaneous. Thus, the installation obviates at least one of the main objections to an exhaust

turbo-driven supercharger fitted on a variable-speed engine.

Water pump and cooling system

As has been mentioned earlier the water pump drive, contrary to common practice, is not combined with that of the fan. Instead the pump is mounted low down on the front of the timing case cover. It is driven at 1·5 times engine speed by the forward extremity of the chain sprocket shaft. The dynamo drive is coupled to the rear end of this shaft.

In general, the water pump is of conventional design. Its cast iron body spigots on to the drive-shaft bearing housing and is secured by six $\frac{5}{16}$ in diameter studs on the timing case. A 5 in diameter gun-metal impellor is screwed on to the left-hand-threaded end of the En 8 driving spindle; the usual spring-loaded carbon ring provides the water seal. Behind the seal is a thrower ring to ensure that if leakage of water should occur it is kept clear of the spindle bearing.

The circulation system is planned to ensure quick warm-up of the water in the cylinder jackets, while keeping the cylinder heads relatively cool. Water drawn from the bottom tank of the radiator is delivered by the pump direct to the cylinder heads through an external gallery pipe. It enters the two heads at six points and is directed by internal ducting on to the six injector sleeves and exhaust valve seats.

Water can return from the cylinder heads to the thermostat either directly or through the cylinder block jackets, and thence the circulation is directed to the top tank of the radiator. These alternative water circuits are controlled by the thermostat valve. When the water is cold, the circulation is largely confined to the cylinder heads; it is not until the cylinder jacket water temperature reaches 170 deg F that the thermostat valve obstructs this circuit and the water flows through the cylinder jackets. The thermostat housing is bolted to and supported by the water outlet connection on one side of the front cylinder head. Another external pipe links it with the outlet gallery on the cylinder block.

An 18 in diameter six-bladed steel fan is employed. It is driven by a $\frac{7}{8}$ in wide, 40 deg V-belt. The driving pulley has an overall diameter of $8\frac{1}{4}$ in. It is keyed on to the forward end of the idler sprocket shaft and secured by a slotted nut and washer. The fan and the $5\frac{3}{4}$ in diameter driven pulley wheel has a ball and roller bearing hub assembly. The rear end of the hub spindle is carried eccentrically in a cylindrical component, which in turn is clamped in the split housing. This housing is formed by a bracket, held down on a face on top of the timing case by four $\frac{7}{16}$ in diameter studs. The eccentric mounting of the fan spindle allows fan belt tension to be adjusted.

Eight rows of flattened tubes with continuous gills form the core of the radiator. Its frontal area is 6·5 ft². The total coolant capacity is 15 gal and the system is pressurized to 4-5 lb/in².

Lubrication system

The gear type oil pump is driven at 0·98 times engine speed by a straight-toothed gear wheel at the front end of the crankshaft. At maximum engine speed it delivers oil at a rate of 27 gal/min. The relief valve blows off at 60 lb/in² pressure. Four $\frac{3}{8}$ in diameter studs and slotted nuts, locked by wire, pull the pump body against the bottom face of the timing case, which is in a forward extension of the sump. It is located by two dowels. Both the pump gears are $1\frac{1}{2}$ in long and have a pitch-circle diameter of 2·360 in. The En 3A driving gear is keyed to a 1·063 in diameter En 8Q spindle, which is carried in $1\frac{1}{4}$ in long steel-backed lead bronze bushes. A 0·997 in diameter, En 32 spindle carries the cast iron driven gear.

Lubricant is drawn through a $\frac{7}{8}$ in diameter pipe from the 9 gal capacity well of the sump, over which there is a gauze strainer of 144 in² area. From the pump, oil is passed by a short pipe and drillings in the side of the crankcase to an external distributor unit, bolted on the front end of the crankcase. Two by-pass valves, a relief valve and a reducing valve are housed in the distributor unit, which also embodies the mounting heads for two oil filter casings. The filters are connected in parallel and have renewable felt elements, through which oil passes from the outside to the inside.

Normally oil goes straight through the distributor unit to a flexible pipe connected to an oil cooler, mounted in front of the radiator. It returns to the centre of the distributor, and thence to the relief valve and filters. The relief valve begins to open when the pressure at the filters reaches approximately 55 lb/in²; it then uncovers a port that releases oil to the timing chain spray. Further increase in pressure opens the relief valve wider and uncovers a larger port to allow surplus oil to be returned to the sump. This oil does not pass through the filters, and therefore filter element life is increased considerably. When the oil is cold its viscosity causes a resistance to its passage to the cooler; if, as a result, the pressure differential between the feed and the return connections to the cooler reaches approximately $12\frac{1}{2}$ lb/in², a valve opens to allow the oil to go directly to the filters.

The distributor unit also embodies a filter by-pass valve. This valve comes into operation only when the pressure differential between the inlet and outlet sides of the filter reaches approximately $12\frac{1}{2}$ lb/in², which is a value such as might be caused by obstruction in the filter elements. After filtration, the oil goes to the pressure reducing valve, the engine main gallery pipe, and the compressor crankshaft bearings. From the pressure reducing valve, oil is fed at 10-15 lb/in² to the camshaft and the tensioner chain.

A $\frac{7}{8}$ in diameter main gallery pipe is employed. It takes the oil to the undersides of the main bearing caps, and thence through $\frac{3}{8}$ in diameter radial ducts to the journals. From the bores of the journals, the oil is led through drillings in the crank webs, as described earlier, to the crankpins.

The low-pressure feed is taken by an external pipe and an internal drilling to the front bearing of the camshaft. Thence it goes through the hollow camshaft to the remaining bearings. As the camshaft rotates, a hole drilled in its centre bearing aligns intermittently with a radial hole in the journal to supply lubricant through an external pipe to the rocker shaft on each cylinder head. Oil returning from the rocker gear lubricates the tappets and maintains the level in the trough under the camshaft. Drillings at the back of the tappet chambers in the cylinder block allow excess oil to return to the crankcase.

MACHINE TOOL LUBRICATION

A 96-PAGE book, entitled "Machine Tool Lubrication," has recently been published by Wakefield-Dick Industrial Oils Ltd., of 67, Grosvenor Street, London W.1. Complimentary copies of this book may be obtained from the P.D. Department of the publishers. Highly complex machine tools are now commonplace and the industry is on the threshold of full-scale automation; also, modern tools are both intricate and costly. Therefore, correct lubrication is of vital importance, and a book on this subject is most useful.

The work begins with a brief description of the machine tool industry; this is followed by a detailed examination of the fundamentals of friction and lubrication. General design features of modern machine tools are discussed at length in another chapter, and then typical tools, such as lathes, milling machines, planers and shapers, are reviewed under separate headings. Finally, the book gives details of a scheme for efficient lubrication in the factory and advice on oil storage equipment, mechanical lubricators and production oils.

PNEUMATIC POWER-ASSISTED STEERING

Clayton Dewandre Compressed-Air Actuated Systems for Large Commercial Vehicles

AIR-PRESSURE actuated, power-assisted steering is well established in the heavy commercial vehicle field. In fact, Clayton Dewandre Ltd. for a number of years have been continuously manufacturing equipment of this type. Since air pressure or vacuum are used to supplement, or even to supplant, the physical effort of the driver, for operations such as braking, gear changing, and actuation of doors, the application of these services to power steering is a logical development. It is obviously an economy to employ this type of equipment rather than to introduce a second power source in the form of a hydraulic system.

At first sight, it might appear surprising that power steering should be considered desirable in this country, since vehicle dimensions and loadings are so restricted by law. However, the inadequacy of a road system in general compels drivers of public service and goods vehicles alike to perform a large amount of low speed manœuvring. This produces conditions in which any reliable measure that reduces driver fatigue is welcomed in the interests of safety and efficiency.

In the commercial vehicle industry there is not complete agreement as to what are the most desirable performance characteristics for power-assisted steering. Nevertheless, there can be no doubt that, as an insurance against power failure, the system must incorporate a mechanical connection between the steering wheel and road wheels. It is important, also, that the power assistance does not destroy or obscure the *feel* that the driver has when steering a vehicle.

When assisted steering is incorporated, the steering box ratio, which otherwise is of necessity rather high for vehicles with very large axle loadings, could be lowered. Power assistance should be used not only to reduce the effort to be applied at the rim of the steering wheel, but also to effect a diminution of the distance over which the effort has to be applied. In this connection, there is every reason for a substantial reduction in the diameters of steering wheels. This would give more space in the cab, and the column could be set at a steeper rake in forward control vehicles. It is surprising that advantage has not been taken of these possibilities. However, the reason is probably that the extra cost of new components is not yet considered to be justified.

True power steering, in which the mechanical connection follows but does not supplement the applied effort, has a completely dead feel. This can be overcome by the incorporation of an artificial feel, as in power-operated aircraft controls. To obtain this effect, the movement of the control can be made to load and unload springs. However, with power steering, as distinct from power-assisted steering, the driver cannot judge the condition of the road surface by the feel of the steering wheel, since the resistance to his turning the wheel is constant for all road conditions and is proportional to the angle turned. The resistance to the steering of the road wheels varies according to the road surface, and the driver should be able to recognize when the surface is slippery, by the consequent reduction in the steering effort required.

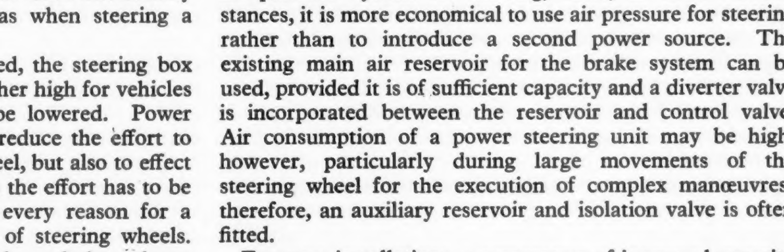
In systems currently employed, control is generally unassisted until a certain minimum effort is applied at the steering wheel rim. Thus, a proportion of the steering effort is always supplied by the driver, and this enables him to obtain the feel of the road conditions. On a completely ice-bound road, steering effort is certain to be below that which brings in power assistance, and so control is effected entirely manually. Also, small deviations from straight-ahead running require no power assistance; in fact, in these circumstances, power assistance would only be an embarrassment. Therefore, in all the Clayton Dewandre installations the control valve is spring-loaded to commence power assistance at a predetermined minimum steering wheel torque.

The systems

Both linkage and steering column type control valves are manufactured by Clayton Dewandre. As has already been mentioned, compressed-air type power assistance equipment is generally adopted if the vehicle already has an air pressure system for braking, since, in these circumstances, it is more economical to use air pressure for steering rather than to introduce a second power source. The existing main air reservoir for the brake system can be used, provided it is of sufficient capacity and a diverter valve is incorporated between the reservoir and control valve. Air consumption of a power steering unit may be high, however, particularly during large movements of the steering wheel for the execution of complex manœuvres; therefore, an auxiliary reservoir and isolation valve is often fitted.

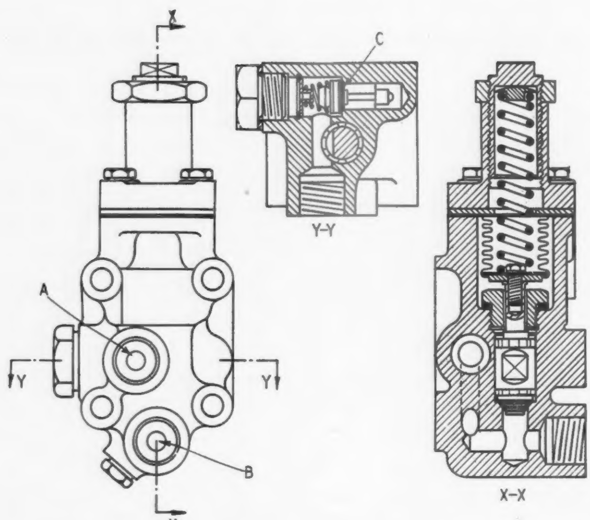
For most installations, a compressor of increased capacity is required when power-assisted steering is incorporated. The Clayton Dewandre compressor is of the reciprocating type with twin cylinders. Three sizes are manufactured, having free air displacements of 15, 10 and 7 ft^3/min respectively at 1,000 r.p.m. In most installations, the compressor is in tandem with the fuel pump and is driven at half engine speed. Also available are a range of three Bendix Westinghouse air compressors with free air displacements of 4, 7½, and 12 ft^3/min respectively at 1,250 r.p.m. In most installations, these are driven at engine speed.

A typical linkage type, power steering system is shown diagrammatically in Fig. 1. The diverter valve, a cross-section of which is shown in Fig. 2, is fitted in the air line



A Main reservoir, B Diverter valve, C Auxiliary reservoir, D Steering control valve, E Power steering cylinder, F Connection to the gear change valve

Fig. 1. In this linkage type, power-assisted steering installation, a diverter valve between the two reservoirs ensures that, if the pressure falls below 60 lb/in², the air in the main reservoir is conserved for brake application only



A *Connection to main reservoir,* **B** *Connection to auxiliary reservoir,* **C** *Non-return valve*

Fig. 2. A diverter valve assembly incorporating a non-return valve

between the main and the auxiliary reservoirs. It prevents the steering system from exhausting the main reservoir.

The operation of the system is as follows. Air is pumped into the main reservoir until the pressure is raised to 60 lb/in^2, when the diverter valve opens to connect the main and auxiliary reservoirs. The pressure in both is then raised to 110 lb/in^2. At this pressure, the unloader valve, incorporated in the main reservoir, opens into an idling circuit through the compressor. The cut-in pressure of the unloader valve is 85 lb/in^2. When fully charged, both reservoirs serve the steering system. Should the air pressure fall below 60 lb/in^2, the diverter valve closes and isolates the main reservoir from the auxiliary one; then the main reservoir can only be used for the brakes.

Components

The diverter valve assembly, Fig. 2, contains a non-return valve. Normally this is seated, but if the pressure in the main reservoir should fall below that in the auxiliary one, the non-return valve opens; thus, all the air is available for the brakes, which naturally are given priority over the steering. If pneumatically-operated doors are fitted in the vehicle, the air supply is taken from the auxiliary reservoir and a non-return valve is not included in the diverter valve unit since, in the interests of safety, the compressed-air circuit to the doors must be operative at all times. If the power steering system is served by the main air reservoir, Fig. 3, a diverter valve is fitted in the pipeline between the reservoir and the steering valve. When the pressure in the reservoir falls below 60 lb/in^2, the diverter valve closes, and manual steering conserves air for the brakes.

A double-acting power cylinder is shown in Fig. 4. The cylinder is an aluminium die-casting, closed at one end and incorporating a bracket drilled and reamed to receive a bush for mounting on the chassis. The opposite end is closed by an aluminium die-cast cover, which is spigoted and bolted to it. A phosphor bronze bush in this cover forms the guide for the En 8 piston rod. This bush is lubricated through a grease nipple on the cover. Two rubber seals, which bear on the rod, are in a counterbore at the inner end of the bush housing. A gaiter is clamped round the cover and the piston rod to prevent dust and water from entering the bush. The piston rod is connected to the steering gear drop arm, a slave arm, or an axle arm.

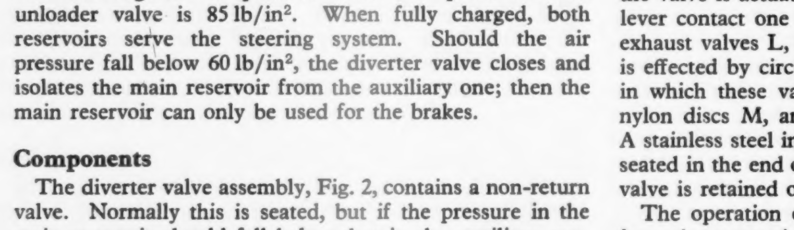
An aluminium die-casting forms the piston. Two synthetic rubber seals are fitted in grooves round its periphery. They are retained by two flanged rings, one on each end of the piston, the flanges projecting between the arms of the U-sections of the seals. The whole seal assembly is secured by bolts passed through holes drilled from end to end of the piston.

Air ports, which are not shown in the illustration, are incorporated at each end of the cylinder. The cylinder bore is 6 in diameter. With an operating pressure of 80 lb/in^2, this gives a thrust of 2,160 lb on one side and 2,050 lb on the other side of the piston.

Shown in Fig. 5 is the linkage type of control valve, which is designed to be fitted in place of the ball joint normally attached to the drop arm. The drag link is bolted to the flange A on the valve body. Ball joint components are housed in the tube B, which is free to slide in the malleable iron housing. Stops at C and D limit the axial movement to 0·062 in ± 0·020 in in each direction. A preload spring E resists movements in both directions. These movements are transmitted by the valve actuating rod F to a saddle block G, which is housed in the actuating lever H. The lever is pivoted at J in an aluminium housing K. When the valve is actuated, adjustable tappets on the ends of the lever contact one or other of the upper faces of a pair of exhaust valves L, machined from En 1A material. Sealing is effected by circular section rings in grooves in the bores in which these valves slide. Return springs, retained by nylon discs M, are held in counterbores by the cover N. A stainless steel inlet valve O, with a hemispherical face, is seated in the end of a central hole in each nylon disc. This valve is retained on its seat by the spring P.

The operation of the unit is as follows. A connection from the reservoir is taken to chamber Q, and chamber R is open to exhaust. Cored passages in the housing and flexible hoses connect the cylinders, S, below the exhaust valve, to the opposite sides of the power piston.

When the steering effort is insufficient to overcome the bias-spring preloading, both sides of the piston are in communication with chamber R and thence to atmosphere through the ports in the exhaust valves. Under these conditions, there is no steering assistance. If the steering effort is increased, one exhaust valve is depressed, and, by seating on the spherical head at the upper end of the inlet valve, closes the exhaust passage from that cylinder to the chamber R. Simultaneously, the head at the lower end of the inlet valve is unseated, and air under pressure is admitted to the power cylinder to give steering assistance. On the opposite lock, the other valve is operated.



A *Connection to compressor,* **B** *Unloader valve,* **C** *Reservoir,* **D** *Diverter valve,* **E** *Steering control valve,* **F** *Power steering cylinder connected to the chassis and drop-arm*

Fig. 3. An installation incorporating a differential steering-control-valve in the column. When a single reservoir is employed, as in this illustration, the diverter valve is fitted in the feed pipe connection to the control valve

Fig. 4. Both the power cylinder and the piston are aluminium alloy die castings. The chromium plated piston rod is carried in a phosphor bronze bush housed in the end cover

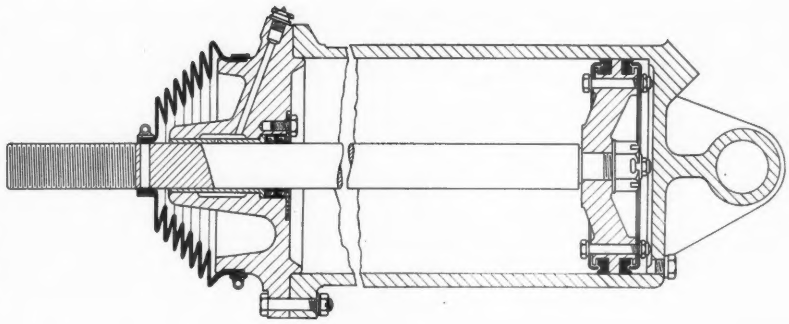
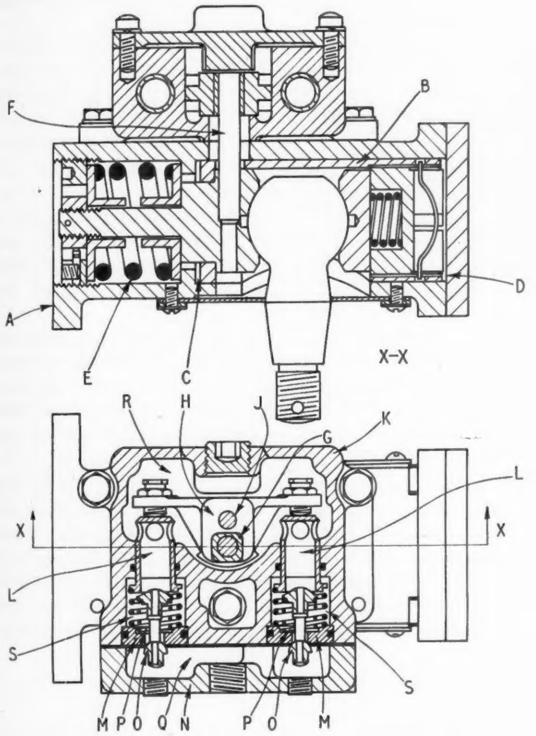


Fig. 5. The ball-pin of the linkage type control valve assembly, which forms part of the drag link, is fitted to the steering drop arm. Axial loads in the drag link are transmitted to the actuating rod and a rocking lever that operates the valves, which are shown in the lower view



The differential type, control valve assembly, Fig. 6, is mounted above the steering box at the base of the steering column. Since rotation of the steering wheel is transmitted to the steering box through a differential gear, and therefore the motion is reversed, the thread of the steering worm has to be of opposite hand to that which would otherwise be required. All the gears have straight-cut teeth. A bronze bush carries the input bevel in the upper part of the split, LM 8WP aluminium alloy housing, and the output bevel is internally splined to receive the worm shaft. The differential cage is freely mounted on a pair of ball races, one on each bevel pinion. Meshing of the gears is adjusted by shims at the joint between two parts of the housing.

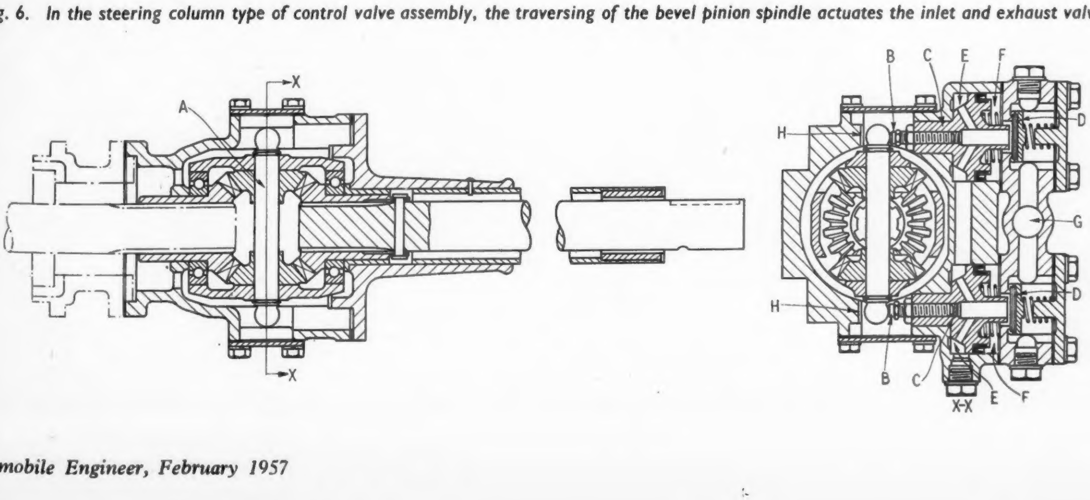
Spherical ends of the differential pinion shaft A bear on a pair of adjustable tappets B screwed into pistons C. The large diameter portion of each piston carries a synthetic rubber seal. In a separate cored casting above the pistons are the inlet valves D. They are made from brass and are faced with synthetic rubber of 90 Shore hardness. A light spring retains each valve on its seat. Below each piston, space E is open to the exhaust and, above it, spaces F communicate with the power cylinder. Air from the reservoir is admitted at G.

Rotation of the steering wheel in either direction causes an angular movement of the bevel pinion spindle, which in turn displaces one of the pistons and unseats the inlet valve. Air under pressure is admitted to the power cylinder to augment the driver's steering effort. When the steering effort is released, retraction of the piston closes the inlet valve and simultaneously releases the pressure, through the axial and radial holes in the actuating piston, to the exhaust passage.

While power assistance is being given, the piston in the active valve is subjected to the pressure in the power cylinder. Hence, a reaction load is transmitted back to the differential gear; since this load is proportional to the assistance obtained, so also is the resistance offered to the driver's effort.

In the event of a pressure failure, stops H prevent damage to the valves. The lost motion obtained in these circumstances is $\frac{1}{8}$ in at the ends of the differential bevel pinion spindle. This represents a free travel of $\frac{3}{8}$ in, as measured at the rim of a conventional 20 in diameter steering wheel.

Fig. 6. In the steering column type of control valve assembly, the traversing of the bevel pinion spindle actuates the inlet and exhaust valves



Mechanization of Body Production

Automatic Handling in the Press Shop and Multi-point Welding for Assembly at the Works of Briggs Motor Bodies Ltd.

IT is an economic axiom that the ultimate aim of production is consumption. Obviously, no useful purpose is served by producing goods unless there are markets in which potential users are able and willing to acquire them, can be persuaded to acquire them or, at least, are not prohibited from acquiring them. In much of the recent comment on automation this fact, either consciously or unconsciously, but nevertheless conveniently, has been ignored. Automation has, in many instances, been regarded as an end in itself that would inevitably and immediately confer substantial economic advantages. The more imaginative speculations envisaged virtually unlimited production of goods in "push-button factories."

No such fanciful illusions were entertained, or even tolerated, at the Dagenham plant of Briggs Motor Bodies Ltd. Although supplying bodies and body components to a large organization, and also feeding overseas assembly plants, always the equipment and methods introduced have been adjusted to meet actual or foreseeable production volume requirements. If full automation had been adopted the productive capacity would have considerably exceeded requirements and plant and equipment of high capital cost would have been used only intermittently. Furthermore, it would have created problems of handling and stacking component panels and of storing assemblies, and thus induced a lowering of efficiency.

Automation is regarded, in a severely practical manner, as a possible means of rendering manual handling and control unnecessary by conveying work between machines or transfer lines; rotating, turning over, or setting up work; and loading and unloading the machines. Handling devices are timed with the machine cycle, and the automatic removal of trimmings is arranged to keep the machines cleared for continuous operation. It can eliminate monotonous, arduous, or hazardous labour in handling work, but will probably require more and better-skilled labour for tooling and maintenance.

The higher the degree of automation the greater are the consequences of the failure of even a small component item of equipment or control. Routine tool changing, machine lubrication, and a general system of preventive maintenance must be carefully organized. Assuming the methods, processes, equipment and controls to have been suitably selected for the work to be performed, the equipment and controls should be simply and robustly constructed and, as far as possible, standardized to facilitate servicing and replacement on the machine, to enable maintenance personnel to be used in any part of the plant, and to simplify the purchase and storage of spares.

The justification for the investment of capital in automation is not the mere elimination or reduction of the labour force. That factor, however, may be of some influence in cases where, for whatever reason, suitable labour is not available in sufficient numbers, or the labour force cannot be augmented to obtain an increased rate of production. The advantage is found in the reduction of the time expended in handling, conveying, loading and unloading the work and the consequent increase in the rate of production. Many machines have necessarily been operated in the past at considerably less than designed speeds as their cycle times were determined by loading and unloading operations. The compounding of reduced handling time and increased

Fig. 1. Press lines laid out for lateral loading and unloading. This method is now being superseded

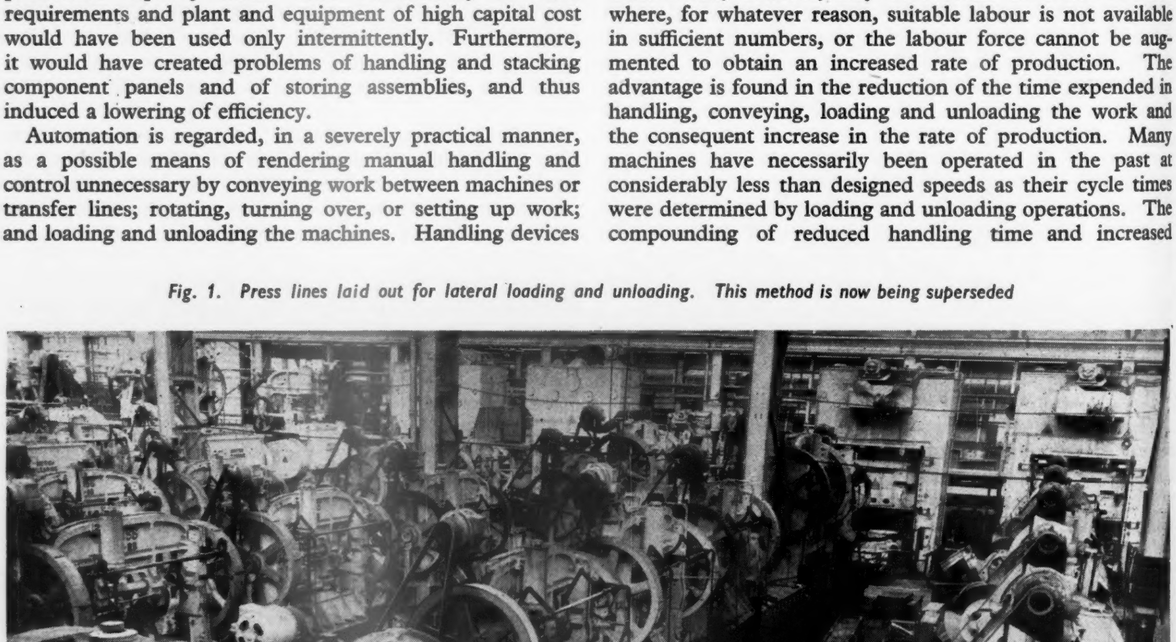


Fig. 2. Current layout of press lines is for through loading and conveyorized transfer

operating speed results in a much higher rate of utilization of costly machines. Other advantages accruing are better utilization of shop space, as room is no longer required for stacking work between machines, and less damage to work by eliminating setting down and picking up between operations.

The trend towards automation will continue but the rate of its application to industry will be dependent on prevailing general economic conditions. Its extension is foreseen as an inevitable consequence of the efforts to reduce the unit cost of production so that consumption of manufactured articles can be increased and industrial activity maintained at a high level.

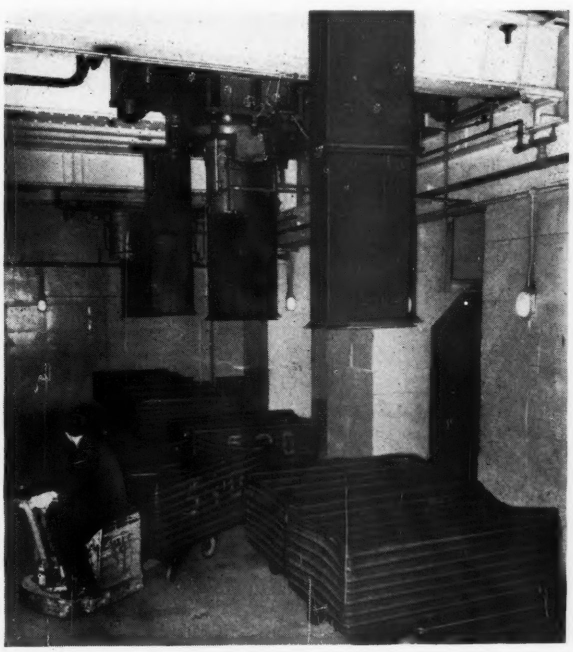
Since 1953, when the company passed into the ownership and control of the Ford Motor Co. Ltd., a large-scale reorganization and development programme has been undertaken at Dagenham. Changes of method or layout have been made, and new equipment has been installed, in a progressing series of steps in order to avoid interruption of production. Although much has been accomplished, the programme is still considerably short of completion. Nevertheless, there is ample evidence in the already operating production lines and in the new tools and equipment to reveal the effectiveness of the reorganization. At every stage the careful planning to obtain enhanced efficiency while maintaining, or even increasing, the flexibility of operation and facility for changeover is apparent.

In the main press shop the outstanding feature is the change in press line layout from lateral loading and unloading, as in Fig. 1, to through loading and transfer, as shown in Fig. 2. The laterally loaded presses were sunk in shallow pits to facilitate work handling and, consequently, the removal of scrap and trimmings was rendered more difficult. Presses on the new lines are joist-mounted and scrap, cut on the press and automatically cleared from the tools, falls by gravity chutes to bins in the basement below. At the present time the wheeled bins are collected by an electric lift truck, as in Fig. 3, but in the fulfilment of the programme the trimmings will fall on to belt conveyors and be delivered directly to an automatic baling machine.

Many of the loading operations are still manually performed in order to retain operational flexibility, but in such cases arrangements are made to relieve the operator of the need to lift or manhandle heavy panels and to ensure his safety. Handling and transfer equipment, which was designed and is mainly built within the Ford organization, can be divided into four main categories; loaders, extractors, turnover devices, and conveyors.

A high degree of standardization has been effected in the design of these units. This is regarded as an essential prerequisite, in order to facilitate purchasing, operating interchangeability, reduction of stores and spares, and ease of maintenance. An important factor is that labour trained to service this equipment can be switched from machine to machine, line to line, or shop to shop with complete familiarity of the component equipment and its problems.

Fig. 3. On the latest lines the presses are joist-mounted and trimmings are chuted by gravity to the floor below



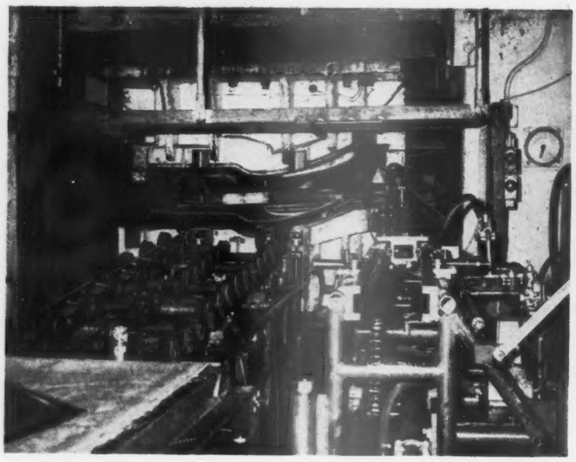


Fig. 5. Side-arm extractor on press in door outer panel line. Panel is raised on the lifters and the side-arm extractor is about to operate

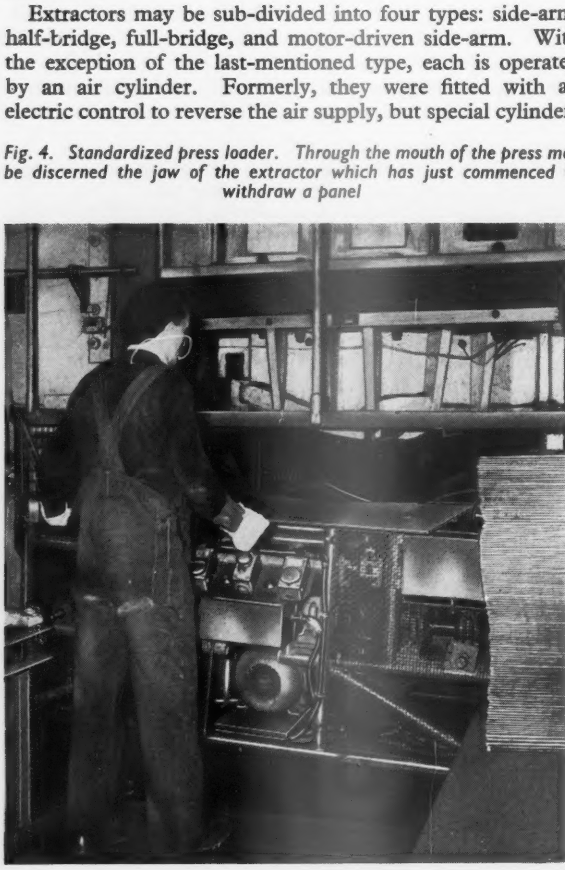
Loaders

All loaders, Fig. 4, have basically the same drive embodying a number of interchangeable component units. An electric motor drives the mainshaft through a reduction gear, and on this shaft is an electric clutch brake providing rapid and efficient stop-start control. A loader for right- and left-hand front and rear door outer panels, and also rear and centre floor pans, has a stroke of 52·5 in and a cycle time of 2·5 sec. Another loader, which in addition to loading also ejects the Consul fender cone through a special wrapping machine, has a stroke of 70 in and a cycle time of 4·75 sec.

Extractors

Extractors may be sub-divided into four types: side-arm, half-bridge, full-bridge, and motor-driven side-arm. With the exception of the last-mentioned type, each is operated by an air cylinder. Formerly, they were fitted with an electric control to reverse the air supply, but special cylinders

Fig. 4. Standardized press loader. Through the mouth of the press may be discerned the jaw of the extractor which has just commenced to withdraw a panel



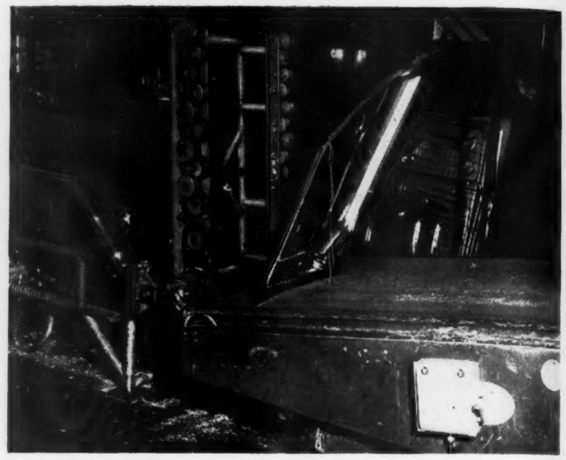


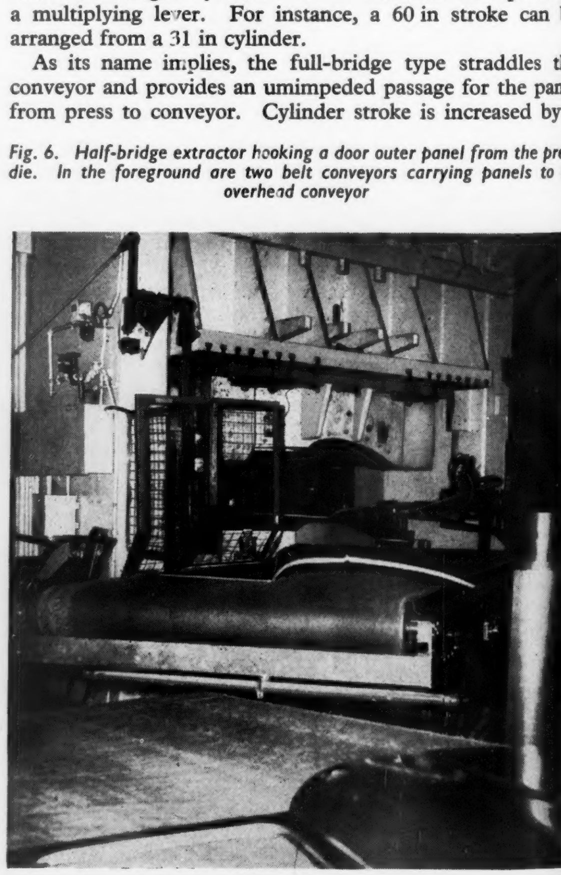
Fig. 7. Standard type of turn-over equipment turning a door panel through 180 deg and delivering on to conveyor

having extended cushion capacity at each end of the stroke are now provided. These obviate the complications and the space disadvantage of the cluster of valves previously required.

Side-arm extractors, an example is shown in Fig. 5, have two carriages, one being directly connected to the ram of the air cylinder. By means of a chain-driven countershaft arrangement, the upper carriage carrying the extractor jaw is given a stroke twice the length of that of the air cylinder. Effective stroke is from 30 in to 40 in. In the half-bridge type, Fig. 6, the extractor jaw is arranged to provide a central draw. A long stroke is obtained from an air cylinder of reasonable length by means of an actuation incorporating a multiplying lever. For instance, a 60 in stroke can be arranged from a 31 in cylinder.

As its name implies, the full-bridge type straddles the conveyor and provides an umimpeded passage for the panel from press to conveyor. Cylinder stroke is increased by a

Fig. 6. Half-bridge extractor hooking a door outer panel from the press die. In the foreground are two belt conveyors carrying panels to an overhead conveyor



chain-driven countershaft system, as on the side-arm units. The pull of the extractor jaw is usually central, but to meet specific requirements variants are provided. For example, on that for use behind the Consul and Zephyr hood top draw die, there are two main carriages, each carrying an extractor jaw which grips the panel at a corner and thus ensures that it is extracted squarely. On this unit an air-operated turn-over is incorporated to invert the panel immediately after extraction.

The motor-driven side-arm type is designed for heavy-duty work, such as the extraction of roof panels and floor pans, and the drive is similar to that of the loaders. It carries a larger extractor jaw and the stroke is 84 in.

Turn-over devices

These air-operated units vary mainly in the form of the cradle, furnished with lines of rubber rollers, which receives the panel to be inverted. The one shown in Fig. 7, will turn a door panel through 180 deg with only an 11 in air cylinder as actuator. On these units also, variant types are built to meet specific circumstances. For instance, a turn-over for a roof panel which runs in a line of presses installed side-by-side is arranged to turn the panel at 90 deg to the direction of extraction.

Conveyors

To link operations, motor-driven belt conveyors are used. These units, Fig. 8, are standardized in three widths, 3 ft, 4 ft and 5ft, and in lengths ranging from 12 ft to 24 ft. All operate at a uniform speed of 50 ft/min. Some units are equipped with an automatic stop which cuts the motor when a panel reaches the end of the belt track. The control can also be arranged to link a series of conveyor units, so that the arrival of a panel at the end of the last conveyor will stop the entire chain.

All these various handling units are mounted on wheeled support framings, enabling them to be rapidly inserted in or withdrawn from a production line. In position, legs are lowered and secured to render the unit immobile. All are adjustable for tilt to enable them to link operations at different heights.

Aids to mechanization

Auxiliary equipments embodied in the dies must be included as factors making possible or conducing to complete mechanization. Reference is made to such items as pneumatic or spring lifters for ejecting the blank from the die, disappearing gauges, trim cutters, and pneumatic kickers.

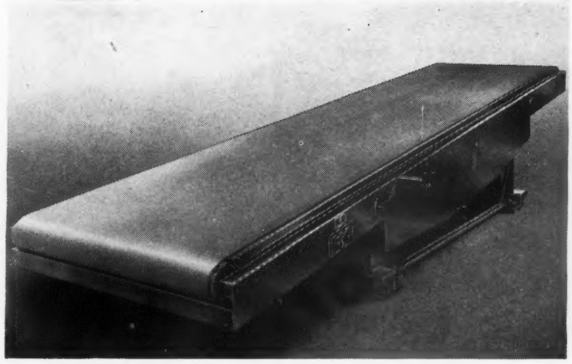


Fig. 8. Belt conveyors, adjustable for height and tilt, are standardized in three different widths

The last-mentioned device can, in many instances, obviate the need for an extractor.

Further items of ancillary equipment are provided or, in some cases, are undergoing development or try-out. These include spray devices to deposit drawing compound automatically on the blank, a hydraulically actuated table to maintain a stack of blanks automatically at a convenient working height, a single-action loader that can be quickly clamped to the end of a conveyor, and a fully automatic blank loader. In all such equipment, design is directed towards simple construction, universality of application, and ease of maintenance.

Typical press line

The door outer panel line may be taken as typical. At the head of the line is a double-action press which is fed with square-sheared blanks. An operator slips a blank on to the loader and depresses two palm buttons. These buttons initiate simultaneously the loader and the press, by way of the electric clutch brake. The pneumatic lifters are lowered before the blank holder bottoms. While the punch comes through, the pneumatic gauges are lowered and the extractor carriage moves in to its forward position. This brings the extractor jaw to a position approximately 0·5 in from the die. As the ram rises, the lifters raise the drawn panel above the ring line, ready to be extracted.

The extractor jaw grips the panel and the extractor carriage moves out, carrying with it the panel. On the initiation of the return stroke the carriage depresses a limit

Fig. 9. Special-purpose wrapping machine for preforming the front fender cone blank

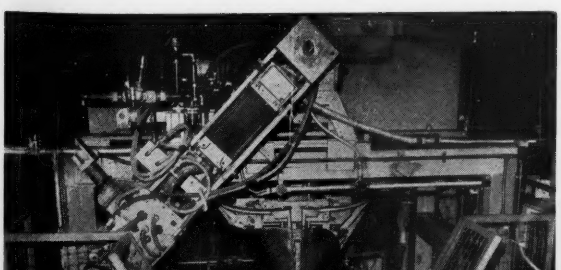
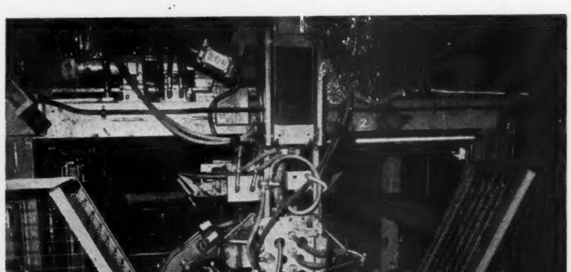


Fig. 10. Automatic spot-welding of front fender cone blank after wrapping operation is completed



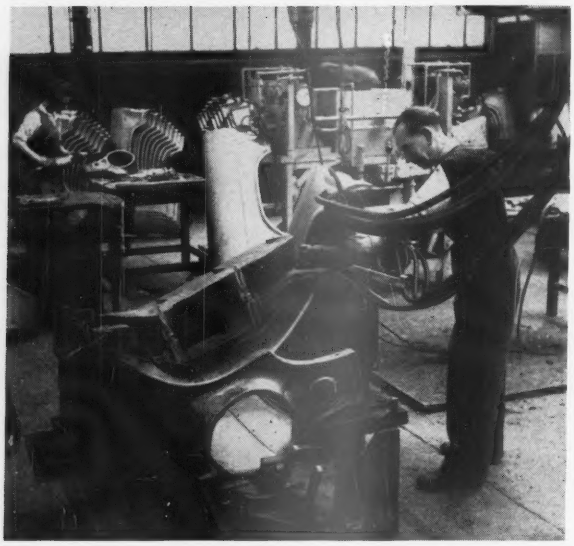


Fig. 11. Hydraulic clinching of fender over reinforcement, and spot welding to tack assembly

switch which breaks the extractor jaw circuit and the panel is released on to the cradle of a turn-over. This device turns the panel through 180 deg and deposits it, right side uppermost, on a 12 ft × 4 ft conveyor.

When the panel arrives at the next press it is manually loaded on to the trimming die. After trimming, the panel is lifted on the upstroke of the press and extracted by a jaw that hooks underneath the panel, to avoid marking its outer surface, and draws it on to a 17 ft 6 in × 4 ft conveyor. These operations are repeated for the flanging of the door margin in the next press and then belt conveyors carry the panel to a point at which it is loaded on to an overhead conveyor. This line is capable of a production of 420 panels per hour. All the presses used are British Clearing models equipped with two-speed "Clearomatic" clutches giving fast approach and slow draw.

Special purpose machines

The complex shape of certain panels calls for special-purpose machines. An example is provided by the wrapping and spot-welding machine, Fig. 9, for preforming the blanks for Consul and Zephyr front fender cones. Reference to the special loader for this machine was made earlier. This wrapping machine is under manual control and, after the blank has been located, depression of palm buttons initiates the cycle. A clamping bar descends to secure the blank approximately centrally from front to rear and then the table hinges upwardly on each side of the clamping bar to fold the blank against the top former. Next, two end gates on the hinged portions turn inwardly to wrap the end of the panel round the end of the former and provide a lap.

An arm pivotally mounted on the front of the machine carries a transformer and four small spot-welding guns. This arm swings down to a vertical position, see Fig. 10, in front of the wrapped end of the blank and, when located, the guns contact and fire to spot-weld the lapped joint. When welding is completed, the guns retract, the arm swings up out of the way, and the machine returns to the ready. The panel is then ejected. All machine operations, in this sequence, with the exception of the welding guns, are hydraulically operated. The scheduled rate of production is 200 fenders per hour.

Controls

A thorough investigation was made of electronic timers for controlling the press and handling sequences. Eventually, however, a mechanical-electric system was adopted, to ensure a positive, mechanical link with the press cycle. A rotary cam box was mounted on each press and driven from the press motion to provide a number of adjustably mounted contacts that would make and break in timed relation to the crank angle of the press. The accuracy of the setting was to 0·1 sec and permitted satisfactorily close sequencing.

More recently the I.G.E. "Masterotor" control has been introduced on certain press lines. In this system a stationary commutator of 180 segments is swept by a brush arm driven directly from the press. Jack plugs are selectively inserted in a board of 180 sockets to bring relays into circuit to initiate and terminate specific operations. This equipment facili-

Fig. 12. Series welding clinch flanges on right-hand and left-hand fenders. The two machines are electrically interlocked

tates and speeds up an initial setting, enables a recorded setting to be rapidly repeated, and allows easy adjustment of a setting to be made to 2 deg of the press crank angle.

WELDED ASSEMBLIES

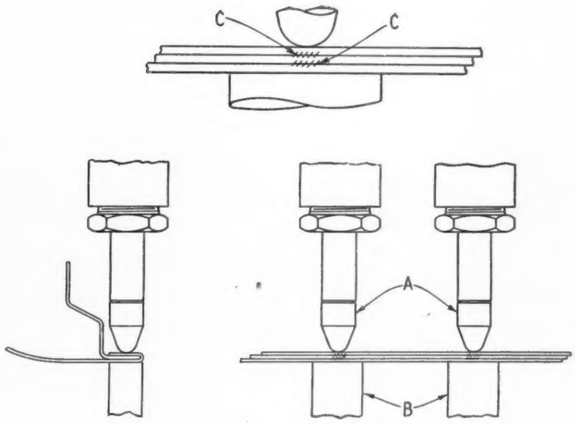
The equipment of the press shop is such that panels, reinforcements, and other components can be manufactured at rates in excess of actual production requirements. Conditions of assembly, therefore, are arranged to allow the most efficient use of labour and storage space to suit the actual build schedule. A high proportion of bodies are exported in knock-down condition and thus panels and sub-assemblies can be produced at a higher rate than is required for the final assembly at the home factory. However, production cannot simply be divided between the two groups as considerations of storage, handling and susceptibility to damage must be taken into account. At the Briggs plant the division is made into four groups which, assuming an output of 30 completed bodies and 20 knock-down bodies per hour merely for example, are as follows:

1. Sub-assemblies to K.D. requirements only—20 per hour.
2. Sub-assemblies that, owing to cubic dimensions or Customs restrictions, are unacceptable for K.D. and are also too bulky for storage in quantity—30 per hr.
3. Sub-assemblies acceptable for both domestic and K.D., but too bulky for storage in quantity—50 per hr.
4. Sub-assemblies capable of storage in quantity—possible production up to press capacity.

Standardization of equipment

The utmost importance is attached to the standardization of welding equipment. It is held that the investment in equipment should be amortized over some period of years and that on occasions within that period re-tooling will become necessary. With well-framed standards established it should be possible to re-use 75 per cent of the equipment in such circumstances.

In the plant the advantages to be gained by standardization are manifold. A much simplified programme of training is required for production staffs and also for maintenance personnel if component equipment is similar both electrically and mechanically and machines differ only in respect of layout. Replacement spares can be reduced to a minimum of types and a single centralized store can feed the entire organization. The anticipated or actual performance can be judged uniformly from shop to shop and consistent results can be expected by the inspection department. The design of tooling is, of course, considerably simplified.



A *top electrode,* **B** *insulated back-up,* **C** *interfacial resistance*

Fig. 13. Diagram of series clinch welding

The high measure of standardization achieved in the multi-point welding equipment at Briggs is impressive:

Presses. Only three types are employed and all are floor-driven. The largest type is the British Federal 84 in $\times$ 186 in $\times$ 30 in stroke, fully hydraulic press, but the two others are of Ford design. One of these is an 80 in $\times$ 80 in machine with 12 in, 18 in or 24 in adjustable stroke, air counterbalanced and hydraulically operated, while the other is an open-fronted, or "C"-type press, 24 in, 36 in, or 72 in wide, with all-hydraulic operation. Hydraulic, pneumatic, water, and electric leads are grouped and fitted with quick, self-sealing connectors. Welding presses are commonly operated at speeds from 200 to 300 workpieces per hour, and are capable

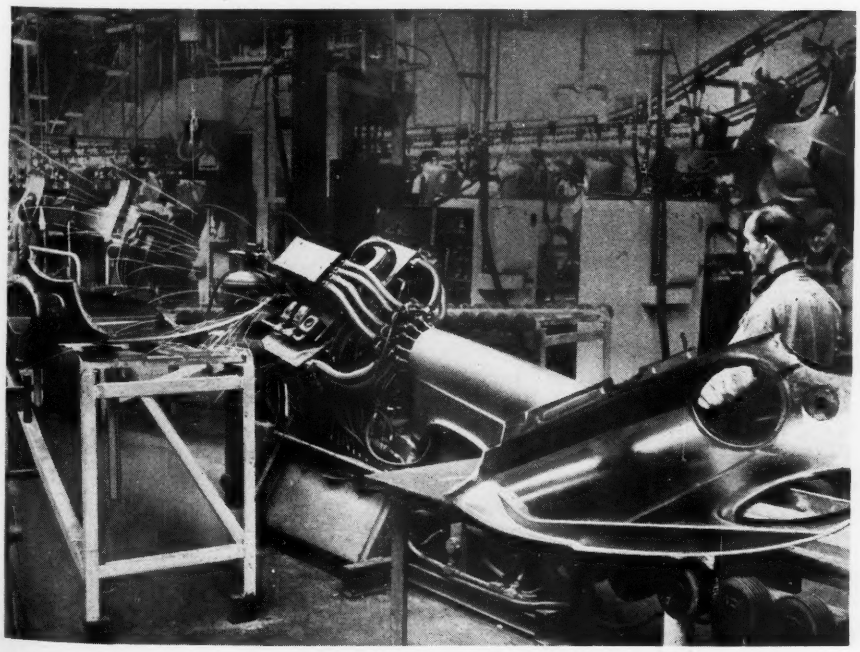
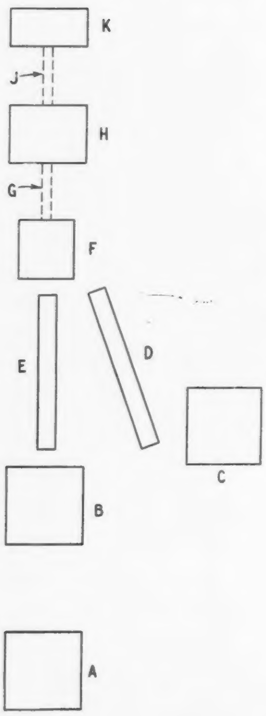


Fig. 14. Welding in headlamp reinforcement to fender. The machine accommodates right- and left-hand components for three different models



A 1st 80 in × 80 in press welder, inner panel, B 2nd 80 in × 80 in press welder, C "C" frame press welder, outer panel, D belt conveyor, E belt conveyor, F pre-clinch fixture, G roller conveyor, H clinch press, J roller conveyor, K final welding machine

Fig. 16. Diagrammatic layout of door assembly line

of development to much higher outputs if required. At such high rates, however, loading and unloading present problems and may be the determining factors in establishing a rate of operation. As an example of high-rate operation, a press cycle time of 5 sec may be quoted for a typical car door.

Transformers. Two types only, both built to Ford specification. For 98 per cent of all multi-point welding equipment a 45 kVA, 50 per cent duty cycle transformer is used. For the remainder a 65 kVA unit is provided.

Welding heads. Two types, of Ford design, each in two sizes having 1·5 in and 3 in strokes respectively. One is of 1·125 in bore and hydraulically operated and the other is of the Savair pattern of 1·75 in bore and tandem air operated.

Control panels. Two types, one for the underbody line and the other for all other multi-point welders. Both use a portable gun weld timer.

Power cubicles. One type. In this Ford scheme, a single cabinet for each phase houses its own ignitrons, thus making possible 3-phase or single-phase operation.

Top electrodes. Standard pattern as used for portable welders.

Bottom electrodes. Two types, both adjustable.

Welding cables. Nine types meet all requirements. All are fitted with terminal connectors and are provided in three different lengths from 12 in to 16 in.

Pipe fittings. A.G.S. fittings are used on all services where applicable. Quick disconnects are provided at all regular uncoupling points.

Hydraulics. Barring those for the underbody line, which are exceptional due to a supply problem, all units are interchangeable. Each carries a pump, motor, valves and one set of spares.

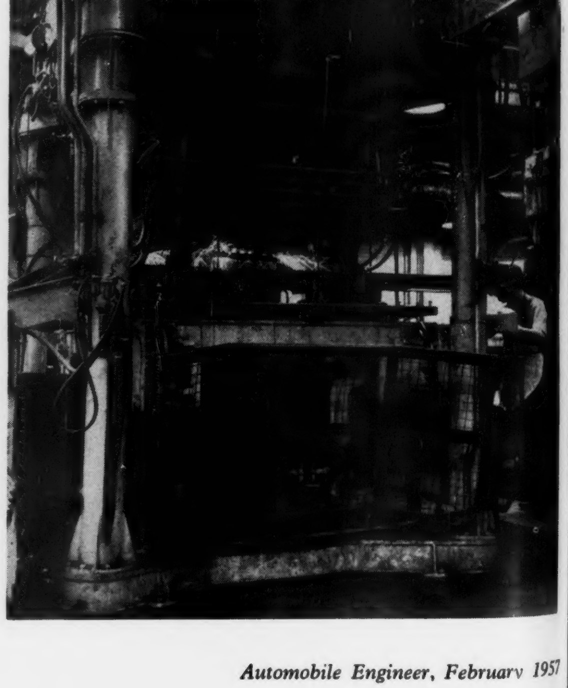
Front fender assembly

This assembly sequence provides a typical example of the use of multi-point equipment for reasons of quality, dimensional accuracy, and accessibility rather than to obtain a high rate of production. A fender is received from the stamping area and loaded in the first assembly tool, Fig. 11, where the rear reinforcement is added. An operator hydraulically clinches the fender over the reinforcement and then, with a portable gun, spotwelds at locations other than the clinch.

Next the fender is progressed to one of a pair of machines, Fig. 12, designed to handle right-hand and left-hand fenders for Consul, Zephyr, and Zodiac models. These machines are wired to a single control panel and the push buttons of one isolate those of the other during the moment of operation. All mechanical actions are pneumatically operated. At this station the clinched flange is series-welded to the reinforcement.

Fig. 15. Assembling air duct sub-assembly to right-hand and left-hand fenders, on "C" frame machine

Fig. 17. 80 in × 80 in weld-press for door inner panel assembly. Push-pull welding is used at some locations



Series clinch welding

In order to minimize the amount of metal finishing required on the outer surface of the fender at this location, it is essential that the surface is not blemished by the welds. Resort is had, therefore, to the method of series clinch welding, see diagram, Fig. 13. By this means the welding current is caused to flow, in different proportions, through each of the three layers of metal from one electrode to the companion electrode of the series pair. The bottom electrodes are of steel and are insulated one from the other.

If the gauge thicknesses of panel and reinforcement are equal, the top layer (clinch flange) will carry the most shunt, the middle layer (reinforcement) somewhat less, and the bottom layer (skin panel) least of all. Assuming a constant weld pitch, any increase in the gauge of the reinforcement will increase its current-carrying capacity and thus will attract more current through the weld between clinch flange and reinforcement. A low flow will be maintained through the skin panel. Since reinforcements are usually of relatively heavier gauge than skin panels, it is necessary only to set the pitch of the welds sufficiently wide apart to bring the drop in resistance of the reinforcement to a par with the interfacial resistance at the clinch flange to obtain a satisfactory result. Actually, on the fender under consideration the outer skin is not even blued. A criterion of good practice is that twice the interfacial resistance at the clinch flange should not exceed the resistance of the clinch flange minus the resistance of the reinforcement. The primary reason for the employment of series clinch welding is to obviate metal finishing operations. While a good weld is required it normally would not be tested to the same standards as structural welding.

Series welding

At the next stage the assembly is fitted with the headlamp reinforcement. An essential requirement at this location is the precise flatness of the face to ensure the correct alignment of the sealed-beam lamps. The special machine for this operation, Fig. 14, is pneumatically operated and can

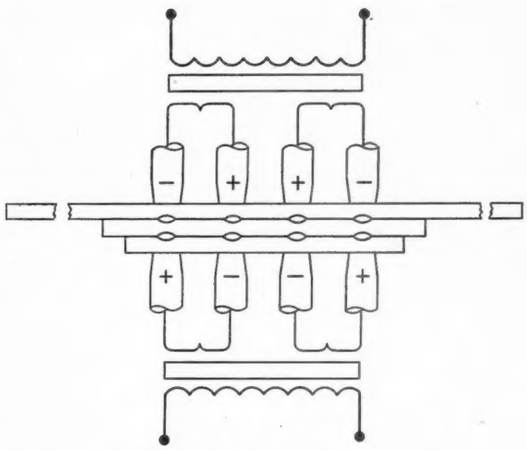


Fig. 18. Diagram of push-pull welding, as used on door inner panel

accommodate right-hand and left-hand fenders for the three models mentioned earlier.

In this instance the series welding is through two layers of metal and it is desirable that they should either be of equal gauge thicknesses or the thinner layer should be against the top electrode. Experience has shown that when two 20 S.W.G. stampings are series-welded at 2 in pitch using a solid bottom electrode of pure copper, approximately 25 per cent of the current is shunted through the top layer, 10 to 15 per cent through the bottom layer, and the remainder through the bottom electrode or back-up block. A good working rule for welds of this type is a minimum pitch of 2 in for the top layer of 20 S.W.G., increasing or decreasing in proportion as the gauge is increased or decreased. As regards the strength of the weld, whether it shows merely a light disc, is of good and smooth appearance, or of indifferent appearance, all are approximately equal.

Fig. 19. "C" frame machine, assembling belt rail reinforcement to door outer panel

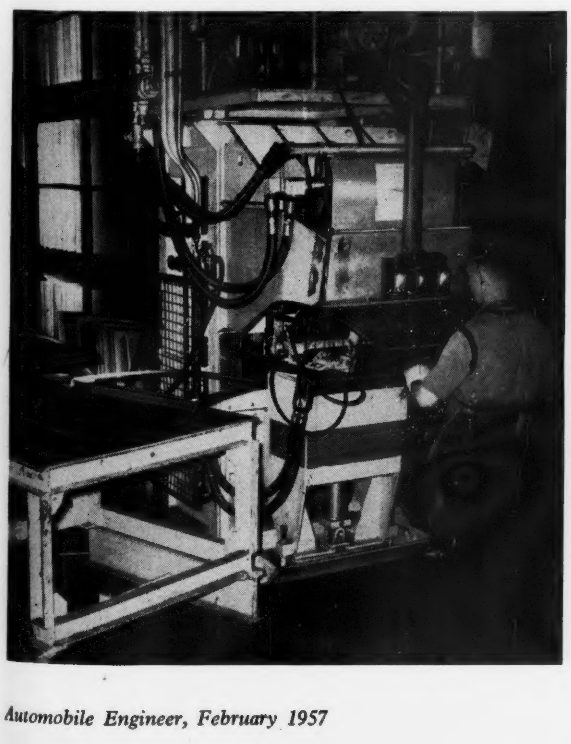


Fig. 20. The end of the line. A door leaving the clinch press for the final welding machine

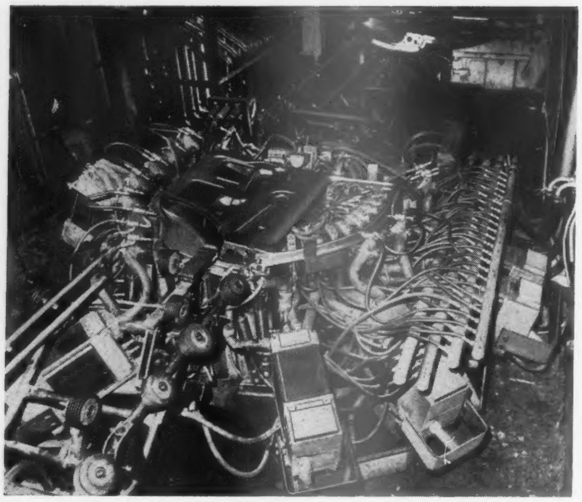


Fig. 21. Table-top welding machine for door assembly

Press welding

The last operation on the fender assembly is to weld in the air duct sub-assembly, Fig. 15. This is done on two "C" frame presses; one for Consul and one for Zephyr and Zodiac components. Press operation is by standard hydraulic equipment. The table is lifted to the operating position and is then hydraulically locked. No mechanical stop is provided. Welding pressure is also obtained from the hydraulic unit, control being by means of a separate valve brought into circuit after the table has been locked. The method of welding is of the same series type as used for the headlamp reinforcement.

On each of the two preceding machines only one welding condition obtains and thus only one weld setting is required. On more elaborate machines, however, it becomes necessary to provide for probable differences in weld time requirement. Accordingly, the standard control panels are designed to allow three timers to be used, one across each phase, and these timers can each be triggered three times. With this arrangement, three different weld times and altogether nine firings over the three phases are possible, giving flexibility and reducing line loading.

The three machines described each have a function additional to welding proper. On the first it is quality—the welds must not blemish the outer surface of the panel. On the second, accuracy—the assembly must be of precise flatness and alignment. The third features accessibility—due to the conditions of approach it would not be practical to produce the assembly by manual means. These machines will not produce more than 50 assemblies per hour, but each is justified by virtue of the additional advantage referred to above.

Side door assembly

A line of machines linked together by conveyors, Fig. 16, producing a complete door from stampings and minor sub-assemblies, provides an example of assembly at a much higher rate than that for the front fender. This line runs all doors for Consul, Zephyr and Zodiac models, the welding units being changed in a similar manner to the dies in the press shop. All welding units are provided with standardized connectors for electric, hydraulic, and pneumatic services and a changeover of one unit for another can be effected in a few hours. Units are serviced and maintained while out of use and a single shift is sufficient to changeover and take trial runs on the next door for running.

Push-pull welding

Door inside panels are assembled in two 80 in $\times$ 80 in press welders, A and B. In the first of these, Fig. 17, the door pillar or the pillar reinforcement, the glass-run channel, the belt reinforcement, and a trim card retainer are welded into the panel. For this work a different type of welding—push-pull welding—is employed. This method is particularly suitable where three layers of metal are encountered or where exceptionally heavy gauge thicknesses need to be welded in locations remote from the edge of an assembly. The system is shown diagrammatically in Fig. 18. Two transformers are used, one above the work and one on the fixture base. It is essential that the electrodes are coupled to arrange the polarity as shown.

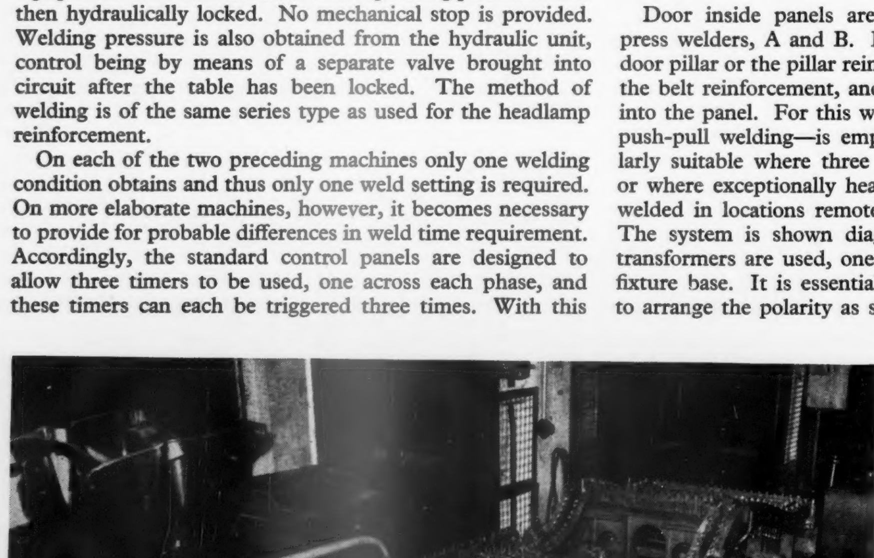


Fig. 23. Underbody fixture trolley in maintenance bay alongside weld-press line. Mounted on the fixture are 588 adjustable bottom electrodes. Five trolleys are in operation while one is being serviced

In press B, the arm-rest support, the remote control tapping plate, and two glass-run brackets are added. These presses are operated by the same standard hydraulic units as the "C" frame presses, but the table rise mechanism is link operated and the weight of the table is mainly overcome by four balancer air cylinders. Control panels and power cubicles are standard "three weld time" units. Actual machine time for this type of press, set on a 24 in stroke, is 5 sec per cycle including weld time. Loading times are additional, and vary from job to job.

Door outside panels are assembled in press C, a "C" frame unit, Fig. 19, which is hydraulically operated in the same manner as the fender line machine. Outside and inside sub-assemblies are transferred on belt conveyors D and E to the pre-clinch fixture F where they are manually fitted together. The pneumatically operated pre-clinch tool forces the two panels together and then bends over the clinch flange at several points to secure the assembly while it is progressed on roller conveyor G to the next machine, the clinch press H. In this an outer ring closes on the assembly to form the peripheral flange at 45 deg, and a central pad then follows to complete the clinch. The assembly is then ejected on to roller conveyor J for transfer to the final welding machine, as shown in Fig. 20.

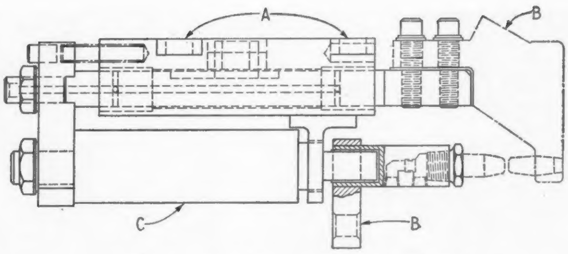
Table-top welding machine

The final welding machine is of the so-called table-top type, which offers advantages in ease of loading and unloading, freedom from restricting top hamper, ease of adjustment and facility of rapid changeover. Incidentally, this trend to reduce overhead equipment in order to free space for conveyors and for travelling crane operation is extending and may later spread to the press shop by the introduction of underdrive presses.

The welding machine, Fig. 21, is built up on a wheeled structure and can be pushed out of the line at a changeover. All the guns are furnished with standard leads and service conduits which are coupled to conveniently arranged common supply lines.

Self-equalizing welding heads

The clinch welds around the periphery are essentially the same as those described on the front fender, and are arranged to avoid blemishing the surface of the outside panel. For the welds between the glass run and the outside panel, however, self-equalizing heads are used to obviate any possibility of disturbing the lay of the marginal flange. As implied by the name, self-equalizing heads are floating units that can assume their own level on an assembly. As will be seen from the drawing, Fig. 22, one of these heads is virtually a complete welder, with both top and bottom electrodes fed. They are employed, provided the welds are adjacent to a trim edge, when panel levels are liable to be somewhat inconsistent.



A fitting lugs, B cable attachment, C tandem air cylinder

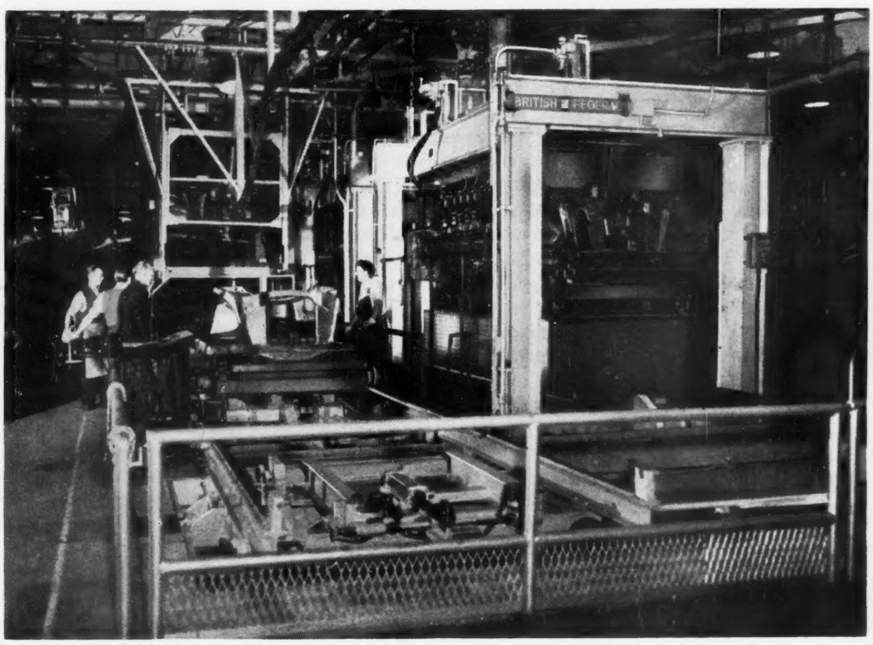
Fig. 22. Self-equalizing welding head

Underbody assembly

For an assembly of this size it was not practical to produce at rates higher than is required for the domestic production programme. In order to make the equipment economically worthwhile, therefore, it was arranged that all model variations could be accommodated and also the underbody sub-assembly in the form it is shipped for K.D. Furthermore, it was decided that whichever assembly was offered to the line, a total of four variants, should be automatically acceptable and that no manual adjustment of the equipment should be required.

The line comprises five British Federal welding presses with, alongside, a return conveyor for the fixture trolleys. On the return line is an unloading station and two loading stations, and at each end is a lateral transfer mechanism. Five trolleys are used and a sixth, complete with welding fixture, as in Fig. 23, is held out as a spare for maintenance

Fig. 24. Underbody weld-press line. Loading end, showing trolley at the second loading station and the transfer mechanisms in position to receive the feed



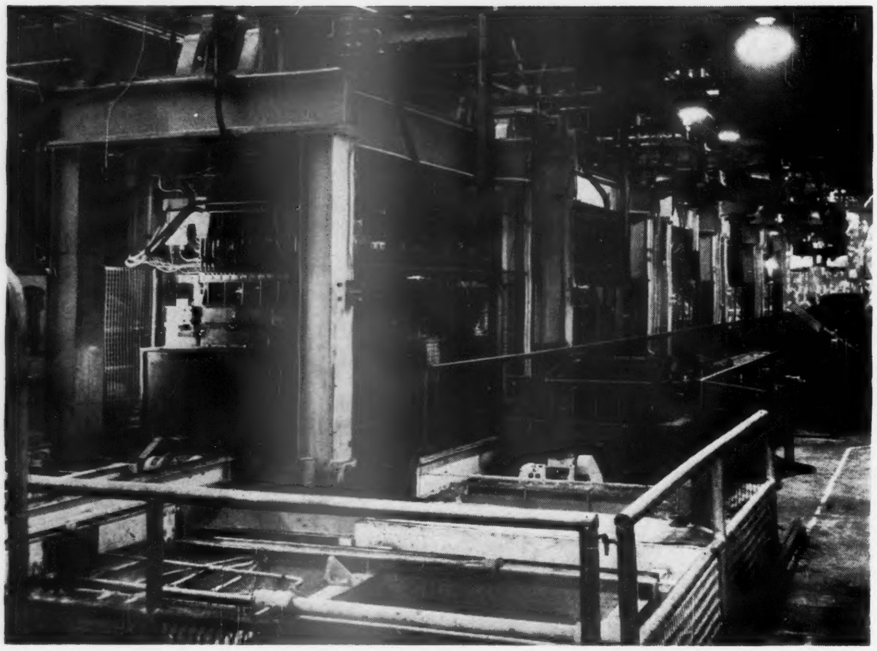


Fig. 25. Underbody weld-press line General view from unloading end, showing trolley on return conveyor

purposes and is changed each week for the next one due for servicing. Fig. 24 shows the commencement of the press line and the loading stations on the left, while Fig. 25 is a general view from the end of the line and showing an unloaded trolley on the commencement of the return track.

All five presses are of identical design; the tooling is varied, of course, to give full weld coverage. As for other presses, the tooling is removable to allow for future model change. Every press is wired for five different weld-time settings that can each be fired twice, giving a total of ten possible firings. The control panels are interlocked so that only one press can weld at a time, thus smoothing the total electrical demand and limiting it to reasonable proportions.

Shuffle welding

With the present set-up, the line is giving 588 welds. To accomplish this number of welds in five presses it was necessary to provide shuffle mechanisms in the tooling wherever possible. Within a given transformer layout, "shuffle" enables welds to be closely pitched by the use of each gun head twice. This is effected, where runs of welds permit, by fitting the gun heads to slides. After the first weld sequence of five shots is completed, the heads are lifted, moved over, pressure is re-applied, and they are then fired through the second sequence. Where the rate of production is not high, as in this assembly line, there is ample time for this method of control to be used. It is adopted to realize the advantage it confers, in terms of equipment, of reduced cost per weld.

True series welding is used at all five press stages. The bottom electrodes on the trolley are all adjustable and are maintained to their correct form by means of templates. A complete weld cycle, from table start to table return, in the heaviest-loaded press can be made in approximately 40 sec. This time may vary slightly due to the welding current interlock action, referred to previously.

To enable production to be continued in the event of a total breakdown of a press in the line, the welding layout is so arranged that the first press and the second press individually effect sufficient welds to all the component parts so that the assemblies may be safely handled. In such circumstances the assemblies can be completed by portable equipment.

Trolley drive

Commencing at the first loading station, the trolley is located on two large, spring-loaded dowels that can be withdrawn pneumatically when required. Aligned with, but laterally displaced from, the medial line of the track rails is a pneumatic ejector cylinder which is electrically synchronized with the dowel withdrawal cylinders. Along the centre line of the trolley is a rack; the first 12 in of which is spring-loaded. Located between the first and second loading stations is an electric motor drive gear, with the final drive pinion aligned to engage the trolley rack. The drive pinion is ratchet-mounted to permit over-run. When the trolley is ejected from the first loading station it travels at a higher speed than would be given by the motor drive. Thus, as contact is established, the rack will either engage directly or the spring-loaded section will lightly jump the pinion teeth and eventually mesh, and the pinion will overrun until it takes up the drive. The trolley is halted by means of spring-loaded brakes on the trolley leading wheels. Operation is by cams located on the track.

When the trolley is finally loaded at the second station, the work is secured by pneumatically operated toggle clamps actuated by the manual application of a loose air line. The trolley is then ejected on to the transfer truck, and from there on is automatically controlled until it reaches the unloading station. Although the hydraulically operated lateral transfer mechanism is mechanically independent of the other elements of the set-up, it is electrically interlocked with the second loading station and the first welding press.

Between each press is a motor drive stage, an ejector cylinder and locating dowels, as already described. Additionally, each press is fitted with a pneumatically operated gathering cylinder. The brakes stop the trolley within 14 in of the dowels, the brakes are released, and the gather cylinder draws the trolley in to the doweling position. On the fast-moving return track, however, there are motor drive stages only; successive drive stages each rotate at a relatively reduced speed to obtain the appropriate rack action.

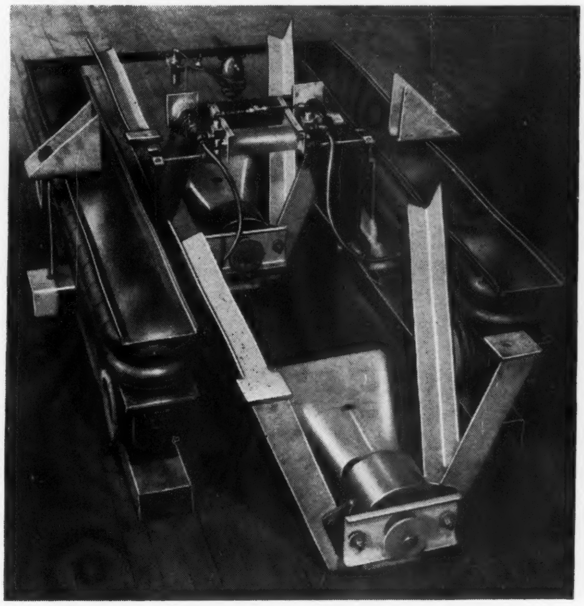
From this line the underbody assembly travels on a slat-type conveyor to the main body-assembly area. All subsequent operations, to the final completion of the trimmed body, are effected manually.

Air Springs for Suspension

Rubber-Bellows Type Springs in Suspension Systems Undergoing Development for British Cars and Commercial Vehicles

RECENTLY, a licence has been granted by the General Tyre and Rubber Co., of Akron, Ohio, to the Andre Rubber Co. Ltd., of Surbiton, Surrey, for the manufacture and further development in Great Britain, of rubber-bellows type air-springs for suspension systems. For several years, the demand for improved suspension systems, particularly for commercial vehicles, has been apparent, but hitherto it has not been possible to improve upon the designs currently employed without substantially increasing costs. However, this new system may well offer a solution to the problem. It has been undergoing development for several years in the United States of America and has already been applied to a large number of commercial vehicles. It is also thought to be suitable for application to private cars; in fact, several of the large American manufacturers are said to be already well advanced in the work of developing it for their projected designs.

Basically, the arrangement comprises simply a rubber bag containing compressed air, which is interposed instead of a conventional spring between the sprung and unsprung masses of the vehicle. An automatic levelling valve is also incorporated to maintain a constant static deflection of the sprung mass, relative to the unsprung mass, regardless of the load carried. This valve has a slight delayed-action, so that as the vehicle rides over irregularities of the road surface, the rubber bellows deflect in much the same way as a conventional spring.



An air spring arrangement for a double trailing axle assembly

Advantages

The advantages of this arrangement are as follows. Its periodicity does not vary with the load. Further, the curve of load plotted against deflection during rapid movements is of exponential form, following the well known law, $P_2/P_1=(V_1/V_2)^{1.38}$, where the symbols P and V refer respectively to the pressures and volumes and the suffixes 1 and 2 indicate respectively the conditions before and after compression. It is a relatively simple matter to maintain a constant height regardless of load, and to trim the vehicle longitudinally and laterally to suit different load distributions. In public service vehicles, this is a valuable asset since it maintains constant the height of the entrance stair. The other advantages of automatic trimming for commercial vehicles are apparent, but perhaps a less obvious advantage for private cars is that headlamp alignment remains undisturbed by changes in load.

Another most important feature is that the available travel between the bump and rebound stops can be used to maximum advantage, and therefore a low spring rate can be adopted. Also, there are no metal-to-metal contacts between the spring and the sprung and unsprung masses. Both these features give good isolation from vibration and noise. Because of the softer springing and reduction in wracking stresses and shock loading, particularly in the unladen condition, the cost of maintaining the vehicle will probably be reduced.

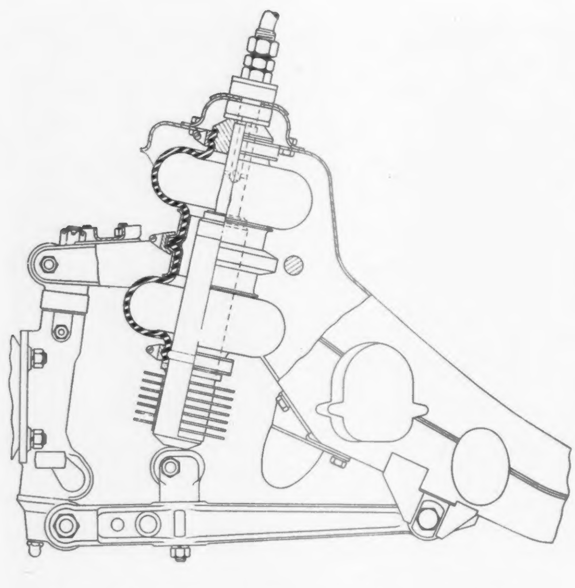
It is also possible that the unsprung mass of systems incorporating this type of spring will be lighter than that obtained with conventional arrangements. This would tend to give longer tyre life and to improve riding comfort. These improvements in ride are important not only to manufacturers of private cars and of passenger-carrying commercial vehicles, but also to the goods vehicle industry, because fragile loads are less likely to be damaged in transit. By comparison with some conventional spring arrangements, the floor height obtainable is low.

Layouts

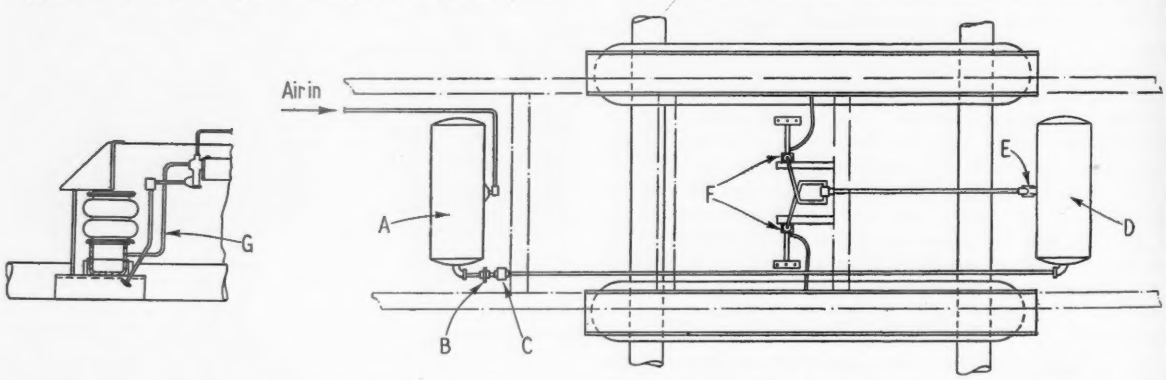
The fundamental requirements of the system are as follows. A source of compressed air is needed: this can take the form of either a compressor or an air bottle. If the supply is not taken from an air bottle, an air reservoir has to be incorporated in the system. A reducer valve is interposed between the air bottle and the automatic levelling valves. A surge tank, which may form part of the structure of the vehicle, can be employed to add to the air capacity of the spring. This has the effect of lowering the periodicity of the system, and in some applications the aperture between the surge tank and the bellows may be restricted to give a certain amount of damping. So far, conventional hydraulic shock absorbers have been found necessary to provide full damping.

Some suggested layouts are depicted in the accompanying illustrations, one of which shows a way in which an air-bellows spring could be applied to an existing, independent front suspension system on a private car. Had the suspension been designed specifically to accommodate the air spring, the bottom pivot of the shock absorber would have been lower. This would have enabled the spring to be secured at a higher level round the shock absorber casing, with a consequent alleviation of the problems of heat dissipation.

In the arrangement illustrated, the spring is fabricated from two cylindrical bellows. Its upper and lower ends are clamped, one to each of the two sliding portions of the telescopic shock absorber. In other words, the installation



Application of an air spring to an existing front suspension system



A *Air-brake reservoir,* **B** *Shut-off valve,* **C** *Feed valve,* **D** *Air spring reservoir,* **E** *Filter,* **F** *Height control valves,* **G** *Flexible connection between height control valve and surge tank*

Layout of an air spring system for a twin-axle bogie of a commercial vehicle. In this particular scheme, two levelling valves are employed

is similar to that of a conventional coil spring and coaxial shock absorber, but its end-fittings are different.

On assembly, the ends of the rubber bag are slid over a rib, of semi-circular cross section, formed round the shock absorber, until they butt against a locating collar, which also is secured to the shock absorber, and which is to prevent the rubber bag from creeping axially. A ring is tightened round the bag to clamp it firmly on the semi-cylindrical section collar, and thus to make an airtight joint.

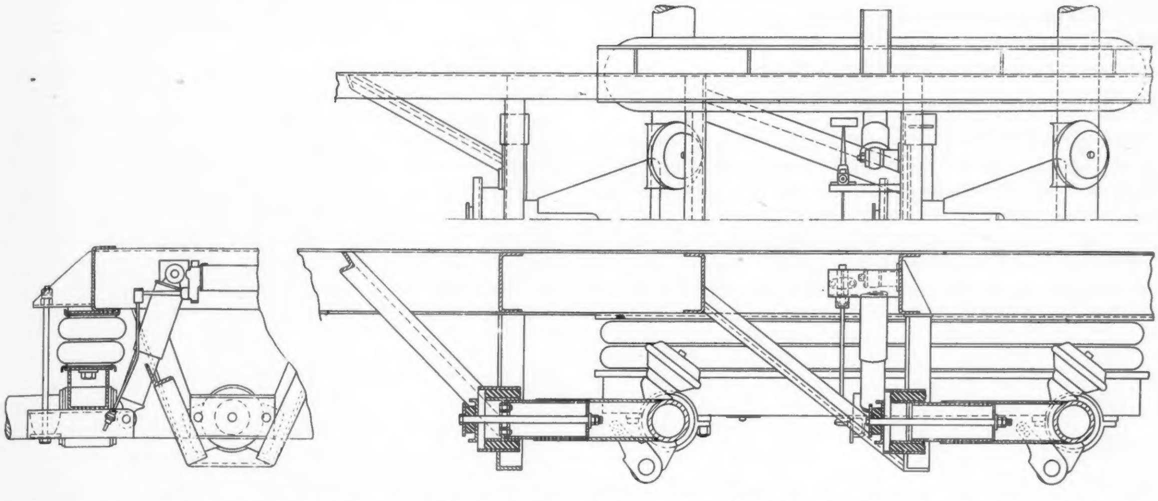
The two pieces that form the spring unit are clamped together at the centre. Their ends are fitted over an inner sleeve and are secured by two outer sleeves and a clamp ring. All three sleeves are of steel. The inner one is of simple cylindrical section with a rib formed round its centre. It is located by this rib, which is held between the flanges at the abutting ends of the two cylindrical rubber bellows. The two outer sleeves are flanged at each end. One is interposed between the abutment flange and the ballooned portion of each of the two pieces of the rubber bellows that forms the spring element. The clamp ring is of V-section, the arms of which fit over the adjacent flanges of the outer sleeves. Thus, when it is tightened, it draws together the flanges of the steel sleeves, compressing the rubber flanges between them. Deflection of the spring is effected by the extension or compression of the ballooned portions above and below this central sleeve assembly. Compressed air is fed into the bellows through a special fitting on the end of the shock absorber. If a surge tank is incorporated, the air supply is taken first into the surge tank and thence to this fitting.

This arrangement, of course, has yet to be fully developed. Possibly the seal of the shock absorber will have to be modified, owing to the fact that it is subjected externally to a relatively high air pressure, which varies as the suspension rides over irregularities of the road surface. Cooling of the shock absorber might also present problems. For this reason it might be preferable to mount the spring and shock absorber separately.

Slightly different damping characteristics might be required to give optimum results. This is because, whereas a steel spring tends to transmit some high frequency vibrations, it is unlikely that an air spring would do so. Therefore, it would probably be desirable to have a shock absorber specifically designed to offer little or no resistance to the small amplitudes of vibration generally associated with high frequencies. Otherwise, the advantage of isolation of high frequency vibrations that is obtained with this type of spring, would not be fully realized. Although air damping is hardly likely to be practicable, it might possibly be used to reduce the load on the hydraulic shock absorber.

One of the commercial vehicle systems illustrated is for a twin-axle bogie. Air is taken from the reservoir for the brake system, through an isolation cock and a feed valve, to the reservoir of the air-spring system. The feed valve is a simple non-return unit, which lifts at a pressure of approximately 80 lb/in². Its function is to prevent the brake system from being rendered ineffective in the event of a sudden demand from the spring system or *vice versa*. From the air-spring reservoir, a pipeline is taken to a T-junction that serves two automatic levelling valves, one for each side of the vehicle. A small filter is incorporated in the outlet union on the reservoir. From each height control valve, a pipe is connected to the surge tank, which is a long rectangular section beam that carries the two axles and forms the lower seating for the long rubber bellows of the spring unit. The upper seating for this bellows unit is on the lower face of the chassis frame side member. Rubber pads inside the springs form the bump stops, and the rebound stops are tie-rods passed through brackets on the upper and lower seating members. A rubber snubber is interposed between the lower bracket and a washer on one end of the tie rod. Another rod, connected to the lower seating member, actuates the arm of the automatic levelling valve, which is mounted on the chassis frame.

From the illustration showing details of the twin trailing axle bogie, it can be seen that rubber bushes are fitted adjacent to the ends of the axles, where bolted-on caps

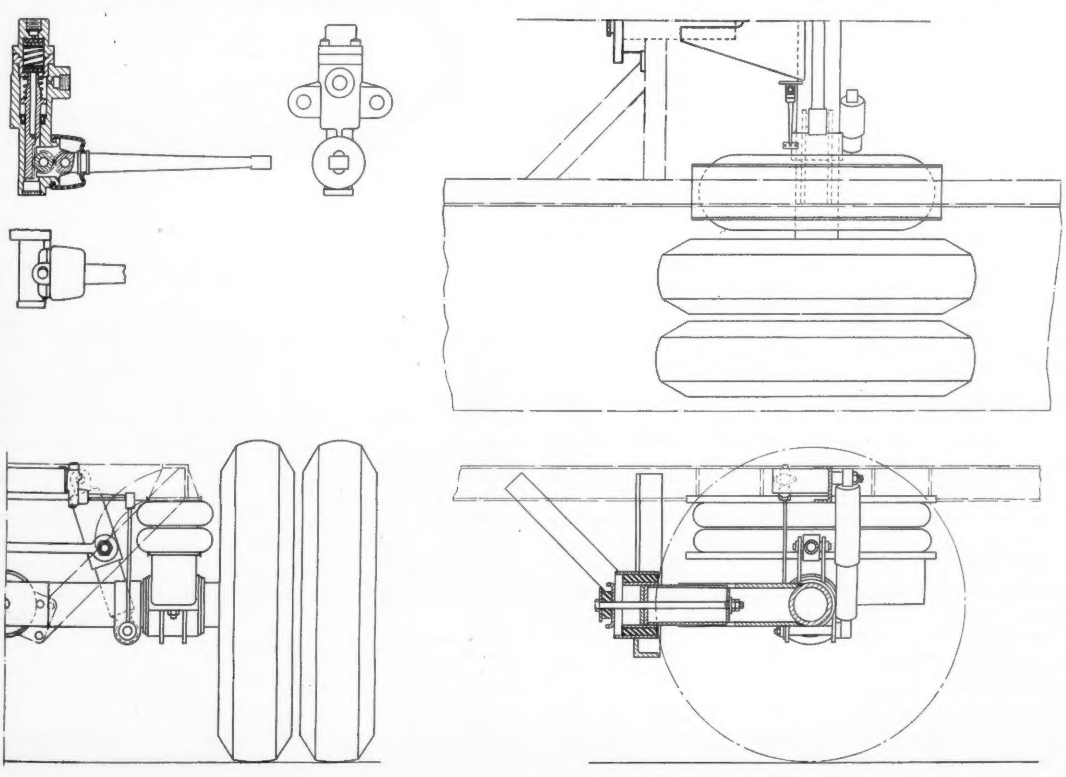


Bellows type air spring layout for two trailing axles. Location is effected by the rubber bushed bracket mounted at the centre of each axle

secure them in semi-circular seatings under the surge tank. These bushes allow for differential angular movement between the axles. Location is effected by two Silentbloc assemblies mounted on forward-extending conical brackets, one on each axle. Each of these brackets, which are mid-way between the ends of the axles, spigots into a rubber bush in a housing mounted on the chassis frame, to locate the bogie laterally. A restricted fore and aft movement is permitted, to allow for the angular motion of the whole bogie about a horizontal lateral axis when one axle deflects vertically relative to the other. This restricted movement is allowed for by the insertion of a circular rubber pad between the bracket on the frame and a washer on the end of the tie rod that retains the axle bracket in its rubber spigot-bearing.

Another illustration shows an air suspension arrangement for a single trailing axle. The arrangement is similar to that for the double trailing axle bogie, already described, except that a Panhard rod is included to afford positive location against transverse movement in the horizontal plane. In a bogie assembly incorporating one driven and one trailing axle, or two driven axles, location would be effected by a Panhard rod, and drag and torque links. These links

A small spring is used with the single trailing axle, and transverse location is effected by a Panhard rod. In the automatic levelling valve, which is shown to a larger scale in the top left-hand corner of this illustration, an idler is incorporated between the rack and pinion

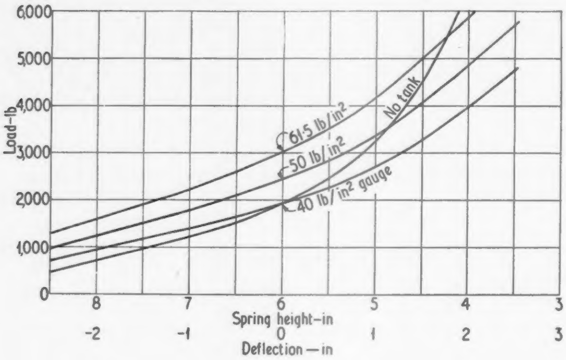


and the Panhard rod would be rubber bushed to avoid offsetting the advantages, with regard to isolation from vibration, gained by the use of the air spring.

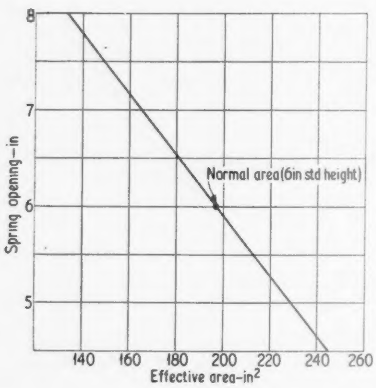
Spring

The spring comprises a natural rubber lining in a case reinforced with plies of tyre cord. In other words, it resembles the inner tube and casing of a tyre, except in that the two are bonded together. An outer layer of Neoprene is applied over the plies. This material is resistant to oil, ozone, deterioration due to temperature changes, and exposure to sunlight. The maximum permissible air pressure in the spring is about 85 lb/in², but there is no low limit except that dictated by the requirements for supporting the sprung mass of the vehicle.

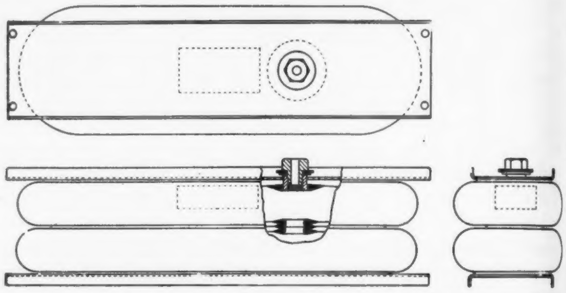
For commercial vehicle applications, the spring comprises two cells, one above the other; its upper and lower faces are bonded to steel channels, which form the seatings for attachment to the vehicle structure. To balance the pressures in the two cells, there are holes in the separating wall. The rate of the spring can be varied by fitting different capacity surge tanks. This has the advantage that the springs can be standardized. Spring characteristics, with and without a surge tank, are given in the accompanying curves and the Table. The spring rate at any point on the dynamic load/deflection curve is given by the tangent at that point.



Above: Dynamic load deflection curves for a small bellows type spring



Left: Curve showing the way in which the effective area of a typical air spring varies with the deflection



Above: Small spring, showing the interconnection of the two cells

Below: A section of an air spring bonded to two steel channels

Air spring	Part number	Effective area in²	Approx. change in effective area in²	Volume of air spring in³	Example tank volumes in³	Example loads lb	Air pressure lb/in² gauge	Air spring without expansion tank: Rate lb/in	Air spring without expansion tank: Frequency c/min	Air spring with expansion tank: Rate lb/in	Air spring with expansion tank: Frequency c/min
$14 \times 8\frac{5}{8}$	211-A-63	48·4	10·8	205	360	1,900 2,450 3,000	39·3 50·6 62·0	944 1,174 1,400	132·5 130·0 125·0	610 720 920	107·0 104·0 104·0
$21 \times 8\frac{5}{8}$	209-A-304	86·7	14·5	460	1,890	4,660 5,860 7,270	54·4 68·3 84·8	1,865 2,270 2,760	118·9 117·0 115·8	1,000 1,180 1,460	87·1 84·5 84·3
$29 \times 8\frac{5}{8}$	210-A-270	129	20·5	660	2,340	5,050 6,300 7,600	39·2 48·9 58·9	2,160 2,575 3,030	121·8 120·0 118·9	1,150 1,350 1,600	89·8 87·0 86·5
$50 \times 8\frac{5}{8}$	208-A-901	251	40	1,260	1,740	5,100 8,700 11,350	20·3 34·6 45·3	2,220 3,330 4,340	124·0 116·4 116·3	1,650 2,450 3,100	107·0 99·8 98·1
$66 \times 8\frac{5}{8}$	209-A-258	354	52	1,690	1,150	10,000 14,700 21,500	28·2 41·5 60·8	4,555 6,190 8,590	127·0 122·0 118·6	3,300 4,600 6,200	108·0 105·0 101·0

To understand properly the function of the spring, it is necessary to appreciate the significance of its fundamental principles, which are as follows. In the static condition, the air pressure, P, acting on the effective area, a, of the spring, is just sufficient to balance the weight of the vehicle. In other words, $P \times a = W$. If the vehicle is then loaded, for example, with passengers, the springs deflect and the air in them is compressed further until the higher pressure acting on the area of the spring again balances the load. At the same time, the height control valve is opened, to allow more air to pass into the spring until the height of the sprung mass returns to normal; nevertheless, if the effective area of the spring were constant, the air pressure in the spring, of course, would remain the same, despite the fact that the quantity of air in it would be larger.

If the loading is effected slowly, so that there is time for any heat generated to be dissipated, in other words, if the compression is effected isothermally, the well known law $P_1V_1=P_2V_2$ applies. Thus, without the height control valve in operation, P_2/P_1 would be equal to V_1/V_2. It follows that if the effective area a of the spring remains constant, $P_2/P_1=a(D_1/D_2)$, where D_1 and D_2 are the spring heights before and after the vehicle is loaded. Therefore, in these circumstances, the spring rate would be constant, like that of a coil or semi-elliptic spring. However, by introducing more air, to maintain a constant static deflection regardless of the load carried, the rate of the spring is increased and, since the natural frequency of the spring is $188/\sqrt{\delta}$, where δ is the static deflection, the periodicity of the system remains constant.

On the other hand, if the suspension is deflected suddenly, for example, in traversing irregularities of the road, compression takes place more or less adiabatically and follows the law $P_2/P_1=(V_1/V_2)^{1 \cdot 38}$. This gives an increasing rate and so tends to prevent resonance of the suspension system. In practice, the action of the spring is further complicated by the fact that the effective area varies with deflection. Whether compression is effected isothermically or adiabatically, the rate of the spring is dependent on the compression ratio. This can be reduced by incorporating a tank to give additional capacity to the spring. The tank can be mounted either directly under the spring, as in the accompanying illustrations, or it can be mounted separately and connected to the spring by pipelines. Some damping can be effected by restricting the passage of air between the tank and the spring. The theory of this type of suspension was dealt with more fully in the March, 1955, issue of *Automobile Engineer*.

Automatic levelling valve

The height control unit is a simple lever-actuated, pneumatic valve. It is possible, of course, that this arrangement will be changed as development proceeds, but the unit currently employed is shown in the accompanying illustration. The lever actuates a rack and pinion, an extension of the rack forming the valve stem, and the complete assembly is housed in a body casting. An idler pinion is interposed between the lever and the rack so that when the end of the lever is moved downwards, the rack moves in the same direction, and *vice versa*. Air enters from the reservoir through the union at the top; the outlet to the spring is in the side of the body casting; and there is a duct to atmosphere drilled axially from the upper end of the valve spindle, the outlet being into the rack and pinion housing.

If the lever is deflected upwards, the valve spindle moves up and lifts the valve off its seat, to allow compressed air to pass into the body of the unit and thence to the suspension spring. This extends the spring until the lever returns to the mean position again and closes the valve. On the other hand, if the lever is deflected downwards, the end of the spindle is lowered until it is clear of the valve. This allows air to escape from the spring, into the body of the unit, and thence through the axial passage in the valve spindle to atmosphere. Again, the action ceases when the pressure in the spring has been reduced sufficiently for the sprung mass of the vehicle to settle to a static deflection such that the lever is in the mean position. As can be seen from the illustration, a non-return valve is incorporated in the union screwed into the body of the unit.

The speed with which the air can flow through the valve, either into or away from the spring, is regulated by metering orifices. The orifice to atmosphere is the small hole drilled from the rack into the axial hole in the valve spindle, while that from the air reservoir is formed by flutes of restricted size round the periphery of the valve plate. This prevents undue wastage of air due to spring deflections while the vehicle is running, and is probably perfectly satisfactory for vehicles equipped with an air compressor. However, it seems probable that if air bottles are used as the source of supply, for example, in private cars, a different type of valve will be needed to minimize loss of air.

GIANT TUBELESS TYRES

In a paper read before the Industrial Transport Association, at the Royal Society of Arts, Mr. F. Easton, a Dunlop service manager, stated that the application of the tubeless principle to giant tyres shows much promise. Further research and experiments are still necessary before sufficient experience can be gained with regard to reliability and roadworthiness. The aims are at reducing the number of deflations caused by simple punctures, improving the resistance to bursting as a result of damage to the casing, saving in cost and weight of the tyre and wheel assembly, simplicity obtained by the employment of a one-piece wheel and one-piece tyre, cooler running and easy repair of simple punctures.

The author stated that there are not any problems with tubeless tyres of up to 6.50 section for use with the car type, well-base rims. However, some of the semi-drop-centre, flat-base rims with loose flanges present special problems with regard to the provision of an effective air seal between the rims and the loose flanges. It would appear that for the majority of sizes up to and including the 9.00—20, the trend will be towards the provision of a tubeless tyre for use with a semi-drop-centre rim with a loose flange; however, for sizes larger than this, more development work is necessary.

It is not practicable to make the tubeless and conventional L-type tyres interchangeable because, although the overall dimensions in the inflated condition would be approximately the same, the rim sizes would be different. This is owing to the fact that the bead seats for the tubeless tyres would have a 15 deg taper as opposed to the 5 deg taper with the wide-base rims and the flat contour of the conventional rims. Therefore, the rim of the tubeless tyre would be bigger and the tyre walls correspondingly shorter. The volume in the well would be such that the total air volume would be almost identical for the different types of tyre. Rims for tubeless tyres would have to have lower flanges to facilitate fitting and removal. Because the ply ratings and air volumes of the tubeless and conventional tyres would be almost the same, so also would be the rated loads and inflation pressures.

AUTOMONOCONTROL TRANSMISSION

Fully Automatic, Electrically-Operated System Developed from the Monocontrol Unit

The electro-pneumatic valve unit and the alternator are mounted, one on each side, at the end of the Wilson type gearbox

In recent years, the preselective transmission has been superseded in A.E.C. vehicles by the Monocontrol unit, in which the fluid flywheel is combined with an air-operated, direct-acting epicyclic gearbox. With this arrangement the gear change lever actuates a small steering-column switch to control electrically the operation of the air valves. Now a further development has been announced. It is the Automonocontrol, which is completely automatic in operation. A transmission of this type will be incorporated in the initial batch of the 850 Routemaster double-deck vehicles, which are to be built by the A.C.V. Group for London Transport.

A great deal of the experimental and development work on the electrical equipment of this system was carried out by C.A.V. Ltd. Noteworthy features of the control are that it is readily adjustable to suit any special conditions of operation, and a control to override the automatic gear changing system can be incorporated. In the new automatic system, speed- and torque-sensitive elements pass signals to the control unit, which integrates these signals to put into effect upward or downward gear changes, as necessary. All the electrical equipment is simple and reliable and has been well proved, so it is expected that virtually no maintenance will be required in service. Since gear changes are made consistently in response to the exact requirements of the vehicle, there may well be a saving in fuel consumption.

C.A.V. fully automatic transmission control

This electrically-operated control system has been specifically designed for application to vehicles already incorporating a Wilson type, direct-acting pneumatic gearbox, but it can also be adapted for use with an electro-hydraulic system of gear operation. The only manual operation the vehicle driver has to perform is the selection of forward or reverse motion.

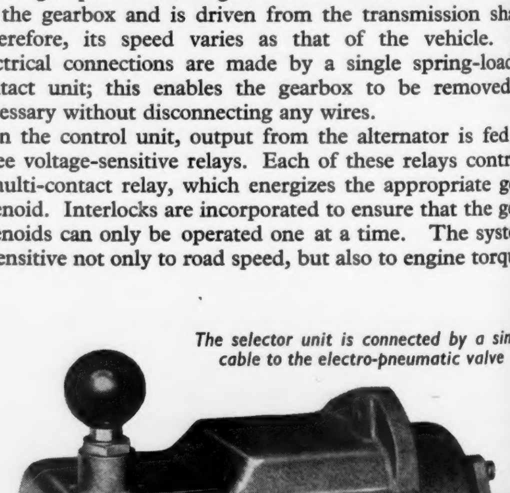
As can be seen from the accompanying illustrations, the equipment comprises several separate components. One is the electro-pneumatic valve unit, which is mounted on the rear of the gearbox, and which is similar to that used with the A.C.V. Monocontrol semi-automatic transmission. It contains a set of five solenoid-actuated pneumatic valves, each of which is connected by a short pipe to a brake-band operating cylinder. The valves are in a common gallery, which is supplied with compressed air from the main reservoir.

A selector unit similar to the change-speed control used in the Monocontrol system is employed. It is a five-position switch, conveniently mounted on the steering column. The five positions are so arranged that the driver can select automatic (A), reverse (R), or hand-controlled first, second and third gears. A spring-loaded plunger prevents accidental engagement of the reverse gear. The unit is connected by a single multi-core cable to the electro-pneumatic valve unit on the gearbox. On vehicles already fitted with the A.C.V. Monocontrol unit, both the electro-pneumatic valve unit and the selector unit can be readily modified for automatic operation.

An alternating voltage output provides the speed-sensitive control signal, which is fed into the control unit. It comes from a speed-sensitive alternator of robust design, incorporating a permanent magnet. The alternator is mounted on the gearbox and is driven from the transmission shaft. Therefore, its speed varies as that of the vehicle. Its electrical connections are made by a single spring-loaded contact unit; this enables the gearbox to be removed if necessary without disconnecting any wires.

In the control unit, output from the alternator is fed to three voltage-sensitive relays. Each of these relays controls a multi-contact relay, which energizes the appropriate gear solenoid. Interlocks are incorporated to ensure that the gear solenoids can only be operated one at a time. The system is sensitive not only to road speed, but also to engine torque,

The selector unit is connected by a single cable to the electro-pneumatic valve

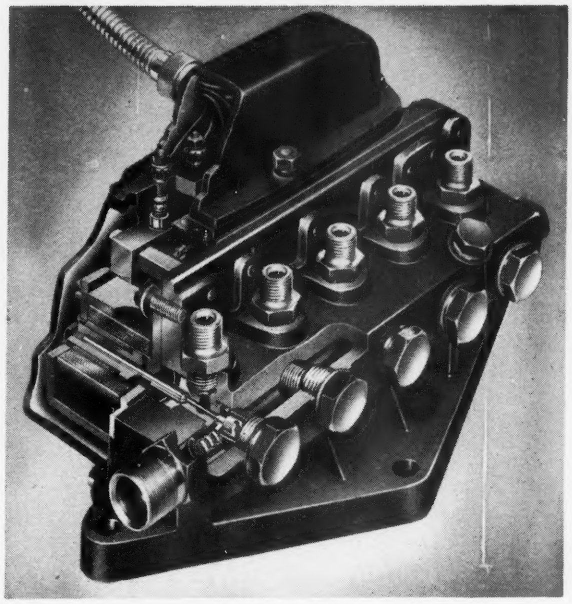


which is, of course, controlled by the accelerator pedal: a lead from the vehicle battery is connected to a switch on the accelerator pedal and thence connections are made to another set of windings on the voltage-sensitive relays. Thus, the speed at which gear changes occur is varied according to the position of the accelerator, and hence the engine torque.

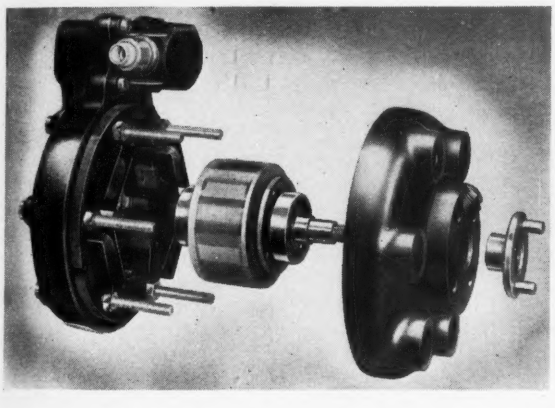
The torque-sensitive accelerator switch is shown in an accompanying illustration. In it, a series of contacts are actuated by the movement of the pedal. These contacts are set in positions corresponding to the one-third, two-thirds and fully depressed positions of the accelerator. In addition, there is a pair of contacts actuated by the initial movement of the accelerator. They are for take-up in a gear and for reversion to neutral when the engine is idling. Yet another pair of contacts is incorporated for the operation of the fourth and third speed interlock for coasting.

Operating characteristics

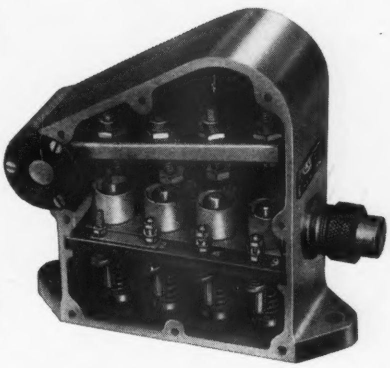
When forward motion has been selected, the vehicle can be driven entirely on the accelerator and brake. The control system automatically changes gear according to the vehicle speed and torque requirements. In any gear, maximum torque is obtained from the engine if the accelerator



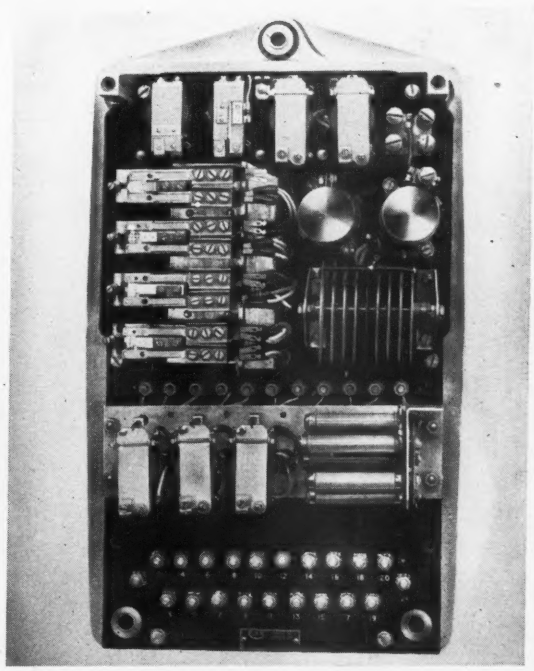
Electro-pneumatic valve unit, containing five solenoid-operated pneumatic valves



Components of the speed-sensitive alternator. A rotating magnet is employed to obviate the need for sliding, brush type contacts



In the torque-sensitive accelerator switch, a series of contacts is actuated by the accelerator pedal



Relays and interlocks in the control unit receive the speed and torque signals and translate them into the gear changes

pedal is fully depressed. Thus, the potential output of the engine is fully realized.

For normal driving, that is, when the accelerator pedal is not fully depressed, changes into higher gears are automatically effected at a speed lower than when the pedal is at or near the limit of its travel. This improves fuel economy and reduces overall wear and tear of the vehicle. Similarly, the speeds at which downward gear changes occur also depend on the accelerator position. This ensures that on level ground, for example, top gear will be held until the vehicle speed drops to a relatively low value. Economy in fuel consumption and reduction of the number of gear changes are, of course, the reasons for the incorporation of these features.

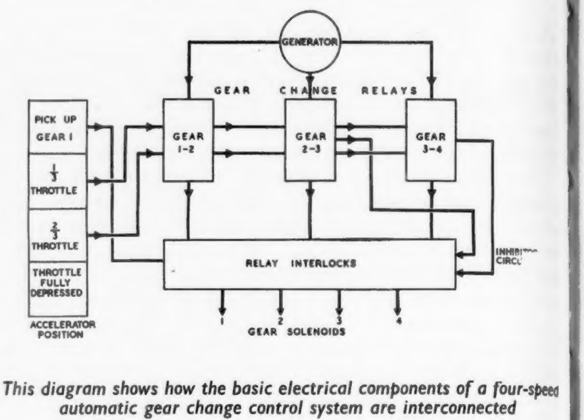
The operating characteristics are shown in the accompanying diagrams. From these, it can be seen that if the accelerator is maintained in the fully depressed position as the vehicle accelerates from rest, the speeds at which upward gear changes are made are much higher than the corresponding values when the accelerator is depressed only one-third of its travel. Similarly, downward gear changes are made at higher speeds when the accelerator is fully depressed. The speed values at which gear changes occur can be simply adjusted to meet operating conditions, so that the optimum settings for any given route can be obtained.

Special features

A number of features, in addition to the basic torque- and speed-sensitive gear-shifting arrangements, have been incorporated in the control system. These features are to meet the operating requirements for public service vehicle applications. With a simple automatic transmission system of this type, when the vehicle is brought to rest the gearbox would remain in first gear and slip would occur in the fluid flywheel. In these circumstances it would be desirable to select neutral manually. To obviate this manual operation, which in practice might not be carried out, the C.A.V. control unit automatically selects neutral in this condition. Thus, fuel economy is maintained. When the driver wishes to set the vehicle in motion, he depresses the accelerator pedal, and first gear is automatically selected; the other gears then follow in sequence.

If a vehicle fitted with a conventional manual gear selector is required to coast at light throttle, the driver normally remains in third or fourth gear until the vehicle almost comes to rest. With a basic automatic system, sequential operation through the gears would be obtained as the vehicle speed decreased. This would result in an unnecessary number of gear changes so, in the C.A.V. system, provision has been made for the vehicle to remain in fourth or third gear when coasting, until a speed of 2-3 m.p.h. is reached; at this speed, neutral is obtained, provided the accelerator pedal is not depressed. Should the driver wish to increase the vehicle speed from the coasting condition, he depresses the accelerator pedal and this causes the control system to select the gear appropriate to the speed and torque requirement.

The basic components that comprise this system can be used to effect automatic control over all forward gears or, if desired, any of the gears can remain under manual control. For example, if the first speed of a four-speed box were required only for use under exceptionally adverse conditions, automatic operation could be restricted to second, third and fourth speed gears, leaving first for manual operation.



This diagram shows how the basic electrical components of a four-speed automatic gear change control system are interconnected

Diagrams

The diagrams of the three-speed and four-speed gear change systems illustrate the interconnection of the basic components. A speed signal from the generator is fed to all the gear change relays. At the same time, another signal is fed to separate windings on each of these relays, which are electrically interconnected; this signal is proportional to the required torque and is determined by the accelerator position. The gear change relays are actuated by the summation of the signals obtained from the generator and accelerator circuits. As the vehicle speed increases, they operate in sequence and simultaneously transmit a signal to the interlock relays, which control the circuit to the gear actuating solenoids in the unit on the transmission casing.

When the vehicle is at rest, no signals are generated and the system returns to neutral automatically. The first gear is reselected when the driver slightly depresses the accelerator pedal and thereby transmits a signal directly to the interlock relays. An inhibitor circuit enables the vehicle to remain in either third or fourth gear when the accelerator pedal is released and the road speed decreases. This circuit cancels the operation of the second to third and third to fourth gear change relays. Neutral is again obtained automatically, when the speed of the vehicle is reduced to 2-3 m.p.h.

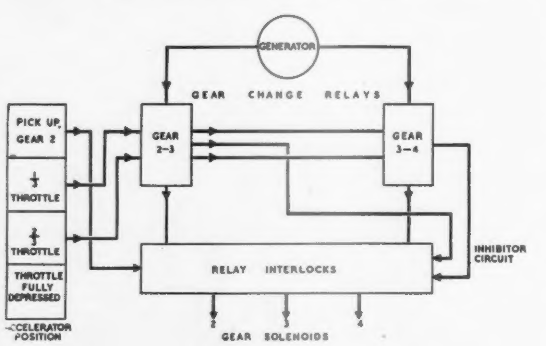
If a vehicle equipped with an automatic transmission system descends a hill, higher gears are selected progressively as the vehicle speed increases. Accordingly, engine braking is not obtained unless some form of manual override is incorporated. With this unit, manual selection can be obtained at will.

An electric interlock, which is cut out of operation when the vehicle speed exceeds about 2 m.p.h., prevents any gear from being obtained manually, unless the accelerator pedal is released. This obviates the severe jerking that might otherwise be experienced if a gear were engaged with the vehicle at rest and the engine running too fast.

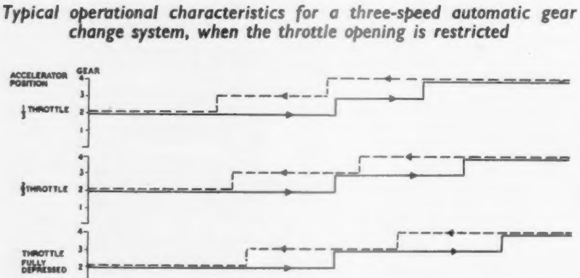
Typical results

Two typical conditions of operation for the system are illustrated diagrammatically. One is a four-speed system, with full throttle sensitivity. The other is a three-speed system with restricted second-to-third gear throttle control, which was incorporated as a result of tests carried out under heavy traffic conditions. It can be seen that with the four-speed arrangement, the road speed at which up-changes occur varies with throttle position. These variations are from 3·5 to 9 m.p.h. for the first change, from 11·25 to 16 m.p.h. for the second, and from 19 to 26 m.p.h. for the

A diagram illustrating the interconnection of the basic electrical components of a three-speed automatic gear change control system

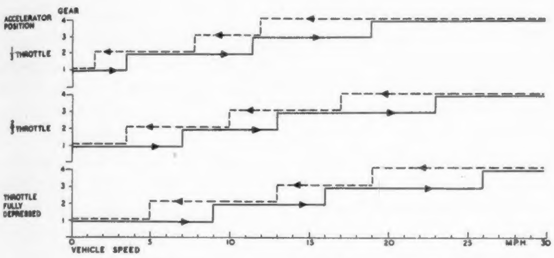


Typical operational characteristics for a three-speed automatic gear change system, when the throttle opening is restricted



fourth gear. The down-changes take place at 12 to 19 m.p.h., 8 to 13 m.p.h., and 1·5 to 5 m.p.h. The difference of speed between up and down gear changes at given throttle positions obviates unnecessary gear changing. With the three-speed box, the first gear change is eliminated and the change from second to third gear takes place at 15·5 m.p.h. for all throttle positions.

Intensive laboratory testing of all the components in the system has been carried out and is still proceeding. The electro-pneumatic valve unit, which is used also in the Monocontrol transmission, was tested earlier and was found



Typical operational characteristics of a four-speed automatic gear change arrangement that is adjusted for full throttle sensitivity

to give a life in excess of one million operations. Further tests have been made to cover the extra duties involved with fully automatic operation.

In general, the aim is at subjecting all parts to tests under simulated road conditions for periods well beyond what is normally regarded as maximum life of the vehicle. Automatic test rigs are in operation day and night, to complete in a few months the operations expected over a period of years. Road testing on a public service vehicle has been proceeding for many months over city and country routes.

Some advantages

The manufacturers consider that the practical value of this control unit has been amply demonstrated. In view of its extreme flexibility, it should be attractive to the operator. Drivers' reaction to the system has been enthusiastic, since it reduces fatigue and enables greater attention to be given to the control of the vehicle. These features are particularly important in dense traffic. Since gear changing is automatically effected, misuse of the gearbox is impossible and shock-loading of the complete transmission system is eliminated. Therefore, an increase in the life of the vehicle transmission components can be expected. A system of this kind is particularly desirable for vehicles of the rear-engine layout, because of the need to relieve the driver of the responsibility for control when he cannot hear the engine clearly.

VIBRATION AND NOISE

Fundamental Data in Respect of the Effect of These Factors on Passenger Comfort

J. L. Koffman, Dipl. Ing., M.I.Loco.E.

CONSIDERATION of the human body as a vibrating structure presents great difficulty because of the different and varying values of the elasticities of bones, muscles, cartilages, tendons and ligaments. Another complication is that muscular tension contributes markedly to the elasticity of the body, which can be generally regarded as a strongly damped resilient mass, the natural frequency and damping factor of which are variable. The natural frequency of a representative relaxed human body on a hard seat is shown in Fig. 1, the mean natural frequency being about 6 c/sec, while the damping is about 0·3-0·5 of the critical damping value.[1]

The transmissibility, or magnification factor, e, is defined as the amplitude of forced vibration/amplitude of the disturbance, or as the transmitted force/impressed force. It is given by:

$$e = \frac{1}{1-(f/f_n)^2}$$

where f is the frequency of the disturbance and f_n is the natural frequency of the system. The value of e has been ascertained by placing persons on a vertically vibrating table and clamping a metal gag between their teeth to establish connection with the skull, the quotients of the amplitudes of the table and skull determining the value of e. In Fig. 2, the range of values of e for ten persons is shown. It can be seen that e decreases as f increases, and the ratio is approximately as the inverse square. The natural frequency is below 15 c/sec. At frequencies in excess of 120, nearly all vibrations are damped out by the body.

Weber-Fechner Law

Sensitivity to weight was first investigated by Weber.[2] In his experiments, weights were placed on the backs of the hands of a number of persons, and it was found that if the weight carried initially was, for example, 100 gr, an increment of less than 35 gr was not noticed; if the basic weight was 200 gr, only an increase in excess of 66 gr was noticed; and so on. This fundamental observation was augmented by Fechner, who carried out some 27,500 weight-difference reaction tests and summarized them in the Weber-Fechner Psychophysical Law, stating that "the increase of stimulus required to produce the minimum perceptible increase of sensation is proportional to the pre-existing stimulus." Thus, if W is the weight producing the sense of pressure and ΔW is the increase in weight that gives a just perceptible increase Δs of the pressure sensation, then:

$$C\frac{\Delta W}{W} = \Delta s$$

where C is a constant, or $s = C \log W$.

The values of Δs vary a great deal. For weights supported on the back of a hand, $\Delta s = 1/2$ to $1/3$, while for a person lifting weights, $\Delta s = 1/50$ to $1/70$. It is of interest that Δs for taste varies between 1/7 to 1/9 in respect of sweetness, 1/6 for saltiness and 1/4 for bitterness. The value of Δs for the effect of sound on the ear is about 1/3, while that for light on the eyes is about 1/100 to 1/120. However, neither the sensation of sound or light follow the Weber-Fechner law exactly. For the comparison of the lengths of two straight lines,[3] $\Delta s = 1/50$ to $1/60$.

Noise

For the measurement of sound intensity, the unit is the *Bel*, which denotes a tenfold increase of intensity, and the *decibel*, *dB*, denoting an increase of 1·26 times, which is the anti-log 0·1. It should be noted that these units represent merely a logarithm of an intensity ratio. Therefore, they do

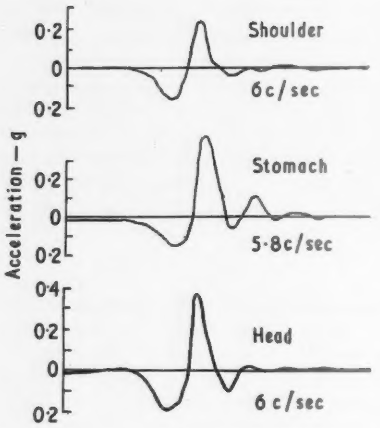


Fig. 1. Some of the natural frequencies of the human body

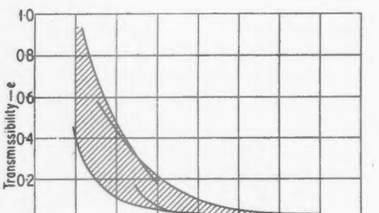
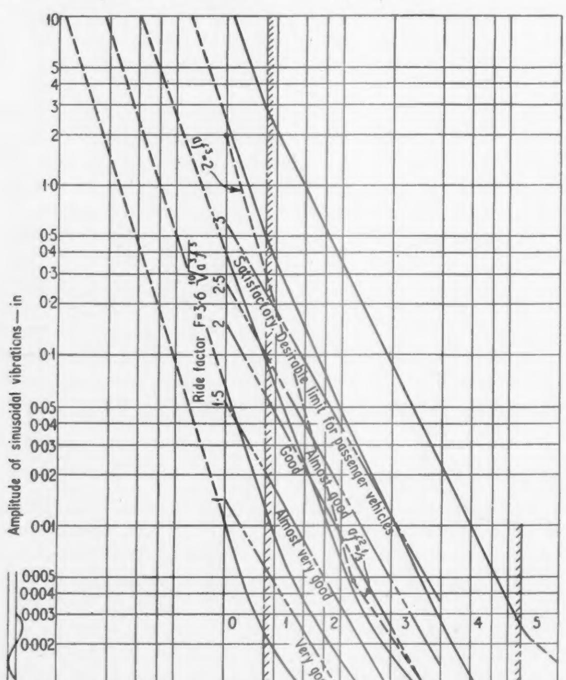


Fig. 2. Transmissibility values obtained experimentally for the human body

Fig. 3. Vibration sensitivity of seated and standing persons

0 *Imperceptible,* **1** *Barely perceptible,* **2** *Distinctly noticeable but not uncomfortable,* **3** *Slightly disagreeable,* **4** *Disagreeable,* **5** *Exceedingly disagreeable*



not constitute a measure of loudness. If I_1 and I_2 are two known values of sound intensity, then the difference of intensity level is given by $\log_{10}(I_2/I_2)$ Bels or $10 \log_{10}(I_1/I_2)$ decibels. One decibel represents an increase in the ratio of 1·26:1, two decibels $(1{\cdot}26{:}1)^2 = 1{\cdot}59{:}1$, three decibels $(1{\cdot}26{:}1)^3 = 2{:}1$, and so on. An increase of intensity of one decibel, that is of about 26 per cent, represents roughly the smallest change of intensity detectable by the ear. The whole range of intensity to which the ear responds is from 1 to 10^{12} or 10^{13}, and in some instances even to 10^{14}. It can, therefore, be expressed as approximately 12 to 13 Bels or 120 to 130 decibels.[4]

Loudness, according to the British Standard definition 2017, is measured in phons. For the commonly occurring frequencies of 500-1,000 c/sec, loudness in phons is nearly the same as the sensation level in decibels, but for all frequencies less than 500 c/sec, the loudness in phons is considerably greater than the sensation level in decibels. The intensity level is given by the number of decibels above a reference level. This is true not only of sound but also of certain other phenomena. In the case of sound, the reference level is that corresponding to an R.M.S. sound pressure of 0·0002 dyne/cm^2 at 1,000 c/sec.

The lowest audible frequency is about 16 c/sec, while the highest varies greatly with the age of the person. At the age of 25 to 35, the highest audible frequency is about 15,000 c/sec, while at 60 to 65 it can drop to 8,000 c/sec. However, the most important range biologically is within 30 to 4,000 c/sec, which embraces human speech, in the widest sense. The ear is most sensitive at about 1,000 to 6,000 c/sec.

Vibration

Extensive tests on the human response to sinusoidal vibrations have been carried out by Reiher and Meister,[5] and later by Meister.[6] A vibrating platform, on which the persons undergoing the tests were either standing or lying down, was employed. When the subjects were in the standing position, they were vibrated first vertically and then horizontally, but when they were lying down, the platform was vibrated horizontally, in the direction of the axis of the body, and then at right angles to it. The frequencies were varied between 1 c/sec and 70 c/sec, while the amplitudes of the vibrations were from 0·0001 cm to 10 cm. A total of 25 persons, from different walks of life, were subjected to the tests.

After being exposed to one mode of vibration for 5 to 8 minutes, the test-subjects were asked to state whether the vibrations were within one of the six comfort zones 0 to 5, defined in Fig. 3, and then the experiment was repeated for a different mode. The results of these investigations were summarized for both vertical and horizontal vibration as applied to standing and recumbent persons. It can be seen from the illustration that the curves showing the limits of the various comfort zones can be represented by the equation $af^x = C$, where a is the amplitude, f is the frequency and x is an exponent varying between 1 and 3. Between about $f = 2$ c/sec and 60 c/sec and $a = 0{\cdot}001$ in to 3 in, the exponent $x = 2$. In other words, the sensitivity depends upon acceleration $b = a(2\pi f)^2 = Caf^2$. If b is constant, $af^2 = C$.

In the range of frequencies $f < 2$, the sensitivity follows the law $af^3 = C$, that is, it depends upon the rate of change of acceleration or jerk j, where j equals $a(2\pi f)^3$. If $a < 0{\cdot}001$ in, the equation $af = C$ applies, that is, the sensitivity is proportional to the velocity of the vibration, v being equal to $2\pi fa$. Thus, within the range that is of interest to designers, the sensitivity, or comfort, is approximately determined by the magnitude of the vibrational acceleration. However, since the responses of different individuals vary and the response of the same individual tested on successive days also varies, the general shapes of the curves are more important than a precise mathematical definition.

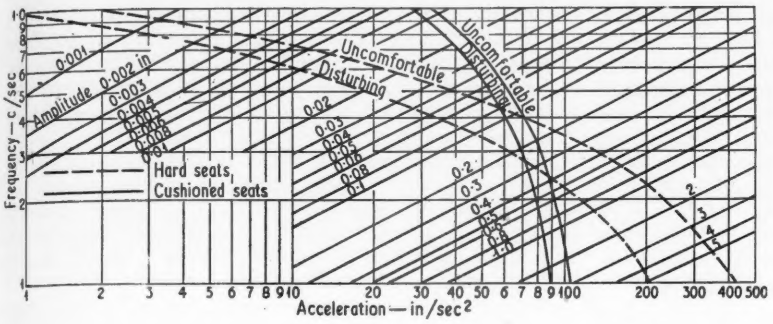
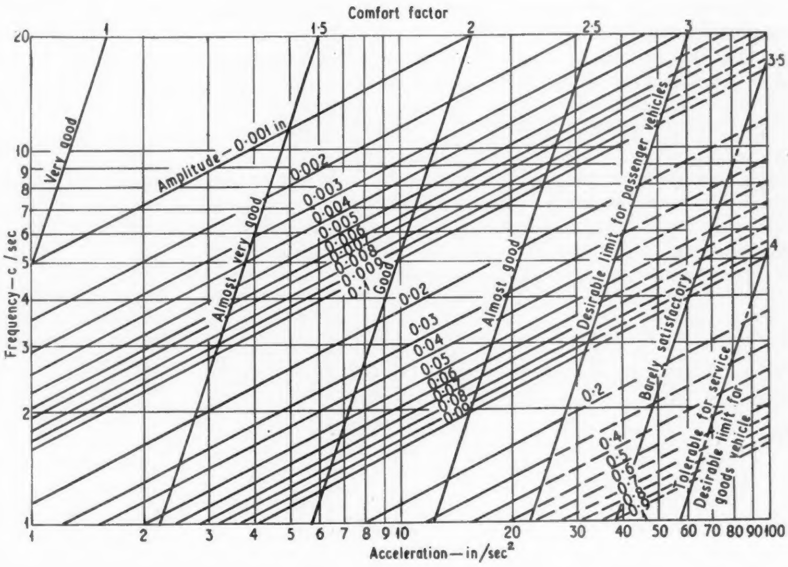


Fig. 4. Experimentally-determined vibration sensitivity of seated persons. A vibration is disturbing when certain organs of the body tend to vibrate at a larger amplitude than the body itself and, to counteract this, certain muscles are brought into play to support them

Fig. 5. Vibration figures of merit, or comfort factors. The twenty-five subjects with whom these tests were carried out were people having some experience with regard to riding qualities of vehicles; this was because their reactions were considered to be more reliable than those of persons inexperienced in this science



Tests to determine the discomfort due to vibrations, as encountered in automobiles, have been carried out by Jacklin and Liddell of Purdue University.[7] The results obtained have shown that vibration sensitivity depends upon vibration acceleration b. It was deduced that for unpadded seats, $b \times e^{0\cdot 6f} = C$, where b is in ft/sec² and f is in c/sec. For vertical vibrations, the condition in which $C=64\cdot7$ is defined as uncomfortable, and if $C=31\cdot2$ it is disturbing. On the other hand, for cushioned seats, $0\cdot13f=C$, where $C=10$ is uncomfortable and $C=8\cdot5$ is disturbing. These limits were explained as follows. A vibration is disturbing when certain organs of the body tend to vibrate more than the body itself and, to counteract this, certain muscles are tightened. It is uncomfortable if the subject feels simply that he wants very little of the treatment. The limits, together with vibration amplitudes obtained from $b=a(2\pi f)^2$, that is, $f=(1/2\pi)\sqrt{b/a}$, are plotted in Fig. 4.

Janeway[8] came to the conclusion that for automobiles, the safe limit for comfort, based on the analysis of most published results, is as follows. Within the limits of $f=1$ to 6, it is given by $af^3=2$, the limiting working value of the jerk being 40 ft/sec³; for $f=6$ to 20, the limiting value is determined by $af^2=1/3$, $0\cdot033g$ being the limit; while for $f=20$ to 60, the limit is set by $af=1/60$, the constant velocity being limited to 0·105 in/sec, Fig. 3.

Other test results, due to Helberg and Sperling[9] are also of interest, although they were carried out to ascertain the effect of vibrations experienced in railway vehicles. Only people with some experience with regard to riding qualities of vehicles were used as test subjects, since their reactions were considered to be more reliable. Some twenty-five employees of the vehicle experimental establishment were put on hard seats, similar to those used in Continental third-class carriages, and were subjected to vertical and horizontal, sinusoidal vibrations. After tests extending from 2 to 10 minutes, their reactions were recorded. The frequency range was 1 to 12 c/sec and the range of amplitude was varied from 0·01 to 2·5 cm. It was found that the limits of the various comfort factors are given by $af^{5/3}=C$, while the actual values of the comfort factor are represented by the equation $F_c=C_c\sqrt[10]{a^3f^5}$, Fig. 5.

Experience has shown that direct application of the results obtained from the rig tests leads to too harsh a verdict on the riding qualities of vehicles. This might be accounted for by the fact that during rig tests, the person concentrates upon the effect of vibrations and is not subject to distractions normally offered during a ride. Extensive additional tests have led to a reduction of the original C_c values by 13 per cent, and it has been found that $F_c=3\cdot6\sqrt[10]{a^3f^5}$, where the units of a, the amplitude, are in inches, and of f, the frequency, are in c/sec. The final results, as shown in Fig. 5, are used as a basis for the evaluation of the riding qualities of rolling stock of the German Federal Railways. Thus, if f is constant, the doubling of b increases F by only about 23 per cent.

These results are replotted in Fig. 3, from which it can be seen that the response to acceleration is very pronounced. Nevertheless, the relationship between response and acceleration is not a direct one, since a given value of b is more unpleasant at low than at higher frequencies. This conclusion is confirmed by practical considerations, since for a

constant value of b, smaller values of f give larger values of a.

It can be seen that the results shown in Figs. 5 and 3 are in contradiction to those plotted in Fig. 4, where the slope of the comfort factor, as determined on the basis of the work done by the vibration, is also plotted. In this case, W=work done=force × velocity. For harmonic motion, the displacement is $x=x_0 \sin\omega t$, velocity $dx/dt=x_0\omega\cos\omega t$ and acceleration$=d^2x/dt^2=-x_0\omega\sin\omega t$. Consequently, $W=m(dx/dt)(d^2x/dt^2)=mx_0{}^2\omega^3 \sin\omega t\cos\omega t=(m/2)x_0{}^2\omega^3\sin 2\omega t$, that is, W is proportional to $mx_0{}^2\omega^3$. While an evaluation on this basis will, for identical values of b, also indicate less discomfort at higher frequencies, the weighting in this respect is more marked.

The equation $F_c=C_c\sqrt[10]{a^3f^5}$ can be used to evaluate the comfort factor for a single sinusoidal mode of vibration or a number of similar vibrations. However, in practice, varying amplitudes and frequencies are encountered in rapid succession, so that to evaluate suspension performance in terms of F_c it is necessary to determine a mean value of the ride factor. While no rig tests have been carried out to determine mean values, experience with the transport of passengers and goods shows that both are far more susceptible to the influence of high ride factor values, that is, large amplitude vibrations, than to low ones. Since both acceleration and the resultant forces increase substantially as F_c is raised, particular consideration should be given to high F_c values. Their effect upon the fatigue of the vehicle components is also important. Comfort factors for composite vibrations can be evaluated as follows:

$$F_{ct}=\sqrt[10]{F_{c1}{}^{10}n_1+F_{c2}{}^{10}n_2+\ldots\ldots+F_{cn}{}^{10}n_n}$$

where n_1 to n_n are the incidences of vibrations having a ride factor F_{c1} to F_{cn} respectively. Thus, if 10 per cent of the vibrations have a ride factor $F_{c1}=4$, and 90 per cent have a factor $F_{c2}=2$, the overall value will be $F_{ct}=3{\cdot}2$.

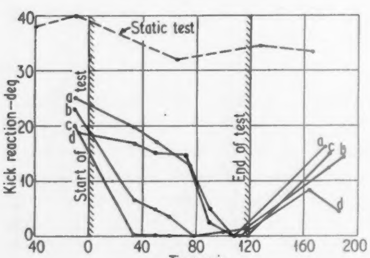


Fig. 6. Effect of vibrations on patella reflex action

Similarly, it is possible to evaluate F_{ct} of vibrations composed of superimposed oscillation of different frequencies applied in the same direction:

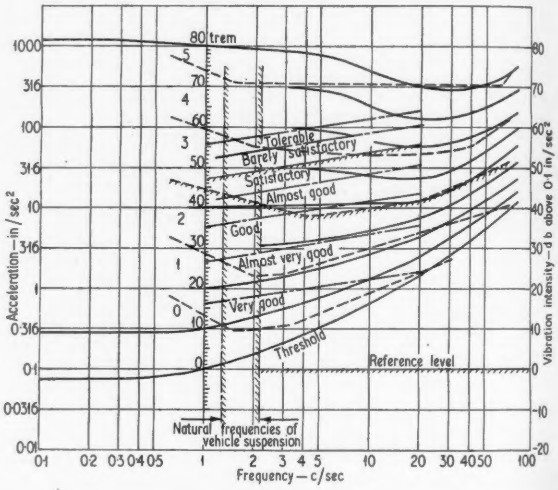
$$F_{ct}=\sqrt[10]{F_{c1}{}^{10}+F_{c2}{}^{10}+\ldots\ldots+F_{cn}{}^{10}}$$

However, in this case, it is essential to quote individual F_c values as well as the F_{ct} values; these will indicate the reason for unsatisfactory suspension performance and the directions in which improvements can be sought.

The data obtained from rig tests, apart from those given in Fig. 5, are not strictly representative of actual operating conditions, since the tests were carried out in stationary surroundings; in practice, the passenger is generally watching a changing panorama, which may affect his reaction; also the different perspective will minimize the effect on him of the motion of distant objects. The variety of noises in a moving vehicle give a subjective sensitivity, to vibration, which is different from that under monotonous laboratory conditions. A more comfortable position may be assumed in a well-upholstered seat. Allowance must be made for the fact that the motions of a vehicle are not purely repetitive; this breaks the monotony to such an extent that single impulses do not produce a feeling of discomfort in proportion to their intensity. Finally, attention is drawn to the fact that the reaction of the persons on test were recorded after a few minutes only, whereas it is often only after 30 minutes to an hour that one may become uncomfortable.

Some useful information has been obtained as a result of patellar reflex tests.[1] Certain nerve ends in the muscles are stimulated by a sharp tap under the knee cap, which causes a sudden movement of the leg. Recordings of kick reactions, expressed in degrees of movement, have shown that the reflex action is appreciably reduced when the person is subjected to vibration; in some cases, it even ceases altogether. The results of tests, in which $f=50$ c/sec and $a=\pm 0{\cdot}01$ in, are plotted in Fig. 6. They indicate that vibrations affect mainly the nervous and physiological reactions, and the nature of the effects can be explained in the following manner.

Fig. 7. Effect of frequency and acceleration on vibration intensity



In assessing the overall effect of vibration on human beings, it must be borne in mind that, to enable a person to maintain an optimum degree of alertness, the body takes protective measures against any form of harmful influence. These measures are partly of a physiological and partly of a psychological nature. The psychological effect is that increases in discomfort automatically give rise to enhanced volition to counteract them. However, the alertness developed by the subject is not proportional to the volition, since the volition generally manifests itself only after a certain stage, and then rises fairly rapidly to a maximum. Before that stage is reached, there is usually even a slight feeling of relief, owing to physiological adaptation to the vibratory conditions. When the stimulus is weak, alertness, or responsiveness, is low, and as the stimuli are intensified, resistance may again become appreciably greater as a result of increased volition, until a point is reached at which the will is adversely affected. The effect of mechanical vibrations within a certain range is thus, in many respects, similar to that of alcohol, a fact also confirmed by tests.[1]

The available data indicate that in the range of frequencies and amplitudes of interest to designers, sensitivity to vibration is to a considerable extent, influenced by acceleration. An attempt is made in Fig. 7, to evaluate in general terms, on the lines indicated by previous work of Postlethwaite,[10] the data of Fig. 3. The acceleration of 0·1 in/sec² at the frequency of 1 c/sec is taken as reference level, and the threshold of sensitivity approximately follows the limit line

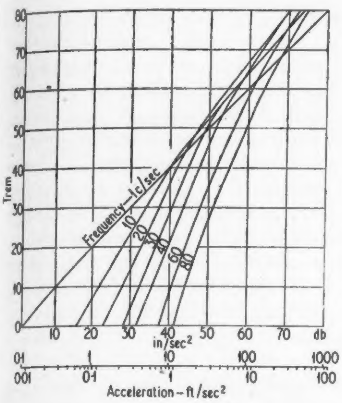


Fig. 8. Relationship between vibration intensity and human sensitivity

of band 1 of Fig. 3. The remaining curves of constant vibration sensitivity are plotted in conformity with available data.

At present there is no commonly accepted term for the units of the curves of constant vibration-sensation. In some German publications, a term suggested by Zeller[11] has been adopted; it is the *pal*, derived from *παλειν*, for swinging motion. One pal is equal to $10 \log 2b^2/f$, where b is in cm/sec² and f in c/sec. However, in English, the term *trem* suggested by Postlethwaite is more widely adopted. The method by which it is derived is similar to that by which the phon is determined. Vibration intensity, like noise intensity, can be expressed in decibels; at $f = 1$ c/sec, decibels and trems are identical. The curves of constant vibration-sensation for other frequencies are also shown in Fig. 7, whilst the relationship between trems and the vibration intensity in decibels is plotted in Fig. 8.

The data suggests that, in general, suspensions should be designed so that the vibration intensity felt by the passengers does not exceed 40 trems, but the system will still be acceptable if a value of 50 trems is not exceeded. If the intensity exceeds 60 trems the riding qualities are intolerable, Fig. 7.

References

1. R. COERMANN: "The Effect of Vibration and Noise on the Human Body," No. GDC.10/7686T. Ministry of Supply, R.T.P./T.I.B.
2. E. H. WEBER: "De Pulsu Resorptione, Auditu et Tactu Annotationes Anatomicae et Physiologicae," Leipzig, 1834.
3. H. ROHRACHER: "Einführung in die Psychologie," 4th Edition, *Urban and Schwarzenberg*, Vienna, 1951.
4. A. WOOD: "Acoustics," *Blackie and Sons*, 1945.
5. H. REIHER and F. J. MEISTER: "Die Empfindlichkeit des Menschen gegen Erschütterung," *Forschung auf dem Gebiete des Ingenieurwesens*, Vol. II No. 11, November, 1931.
6. F. J. MEISTER: "Die Empfindlichkeit des Menschen gegen Erschütterung," *Forschung auf dem Gebiete des Ingenieurwesens*, Vol. VI, No. 3, May/June, 1935.
7. H. M. JACKLIN and G. J. LIDDELL: "Riding Comfort Analysis," *Eng. Bull. Purdue University*, Vol. XVII, No. 3, May, 1933.
8. R. N. JANEWAY: "Vehicle Vibration Limits to Fit the Passenger," *S.A.E. Preprint*, March, 1948.
9. W. HELBERG and E. SPERLING: "Verfahren zu Beurteilung der Laufeigenschaften von Eisenbahnwagen," *Organ. für die Fortschritte des Eisenbahnwagens,"* Vol. XCVI, No. 12, June, 1941.
10. F. POSTLETHWAITE: "Human Susceptibility to Vibration," *Engineering*, January 28th, 1941.
11. W. ZELLER: "Masseinheiten für Schwingungsstärke und Schwingungsempfindingsstärke," *A.T.Z.*, Vol. 51, 1949.

Chevrolet Petrol Injection

The Ramjet System, Which is Based on the Continuous Injection Principle

THE Chevrolet Ramjet fuel injection system is entirely new in principle. It controls accurately the air and fuel ratio of the mixture supplied to each cylinder of the engine. The unit is an optional extra for the 1957 Chevrolet cars and is being manufactured by Rochester Products Division of the General Motors Corporation. Advantages of the system are said to be rapid response to the accelerator control and smooth running during the warm-up period when starting from cold, and also under normal operating conditions.

In principle, the operation of the system is simple. The accelerator pedal controls the volume of air admitted to the engine; a mechanism continuously measures the volume of this incoming air and automatically meters the precise quantity of fuel to be mixed with the air. Other mechanisms are incorporated to enrich the mixture for acceleration, hill-climbing and warm-up, and also to ensure immediate delivery of fuel to the nozzles for starting, to provide for smooth idling, and to cut off fuel when the vehicle is coasting downhill.

Specially designed components, such as air-metering and fuel-metering systems, and fuel nozzles, are included in the system. On the Chevrolet engines equipped with conventional carburettors, a one-piece cast iron intake manifold and engine cover are used, but when the fuel injection unit is installed, this cover is replaced by two separate aluminium castings. The lower casting forms the top cover of the engine, while the upper one contains the air passages and incorporates the base for the air-metering and fuel-metering systems. Other new components include an auxiliary fuel filter, a special ignition distributor with a flexible drive to actuate a high-pressure fuel pump, and a new electric control system for starting the engine initially when it is cold.

Air and fuel intakes

Air for the engine passes through a cleaner into an air meter, and thence to the intake manifold, cylinder head, and combustion chamber. The entrance to the air meter is a venturi. As the air is drawn into the engine, there is a tendency for a vacuum to be created in a pipe connection between the venturi and the fuel meter. The degree of vacuum is an accurate measure of the volume of air being drawn into the engine.

The fuel meter is used to regulate the supply to the injection nozzles. From a conventional, engine-driven pump, the fuel is passed through a fine filter and into a reservoir in the fuel-meter housing. The quantity of fuel in this reservoir is maintained at a fixed level by a float-controlled valve. Another pump, submerged in the fuel reservoir and actuated by the flexible drive from the ignition distributor, passes fuel at high pressure into a central passage, where it must lift a ball check valve before flowing through a series of small holes into a metering chamber. At this point, the fuel can go either to the injection nozzles at the intake ports or back to the reservoir, depending upon the position of a plunger. When this plunger is raised, fuel flows back to the reservoir. As the plunger is lowered, an increasing proportion of the fuel flows to the injection nozzles and the remainder returns to the reservoir. The ball check valve in

the central passage permits fuel to flow from the pump only when the fuel pressure exceeds about 15 lb/in², so that any vapour that may have formed is compressed to liquid form.

As the incoming air passes through the venturi and is measured, it sends a vacuum signal to a main control diaphragm above the plunger in the fuel meter. Depending upon the degree of vacuum, the diaphragm meters fuel by raising or lowering a lever that actuates the plunger. Thus, the precise quantity of fuel required for the volume of air being used by the engine is delivered with great accuracy. All the levers in the fuel-metering system are counterbalanced so that their movements are unaffected by their own weight; their positions are determined only by forces exerted by the pressure sensitive diaphragms.

The intake manifold incorporates eight individual passages, or ram pipes, one for each cylinder. Plastics insulators carry the fuel injection nozzles in the lower part of the intake manifold, near the cylinder head induction ports. A throttle valve, controlled by the accelerator pedal, determines the quantity of air and, as previously explained, the quantity of fuel supplied to the engine.

In order that the amount of fuel injected shall be determined solely by the fuel-metering system, and shall not be influenced by variations in vacuum around the nozzles in the ports, the nozzles are designed to inject fuel against atmospheric pressure at all times. This is accomplished by supplying air from the cleaner to a small chamber in each nozzle.

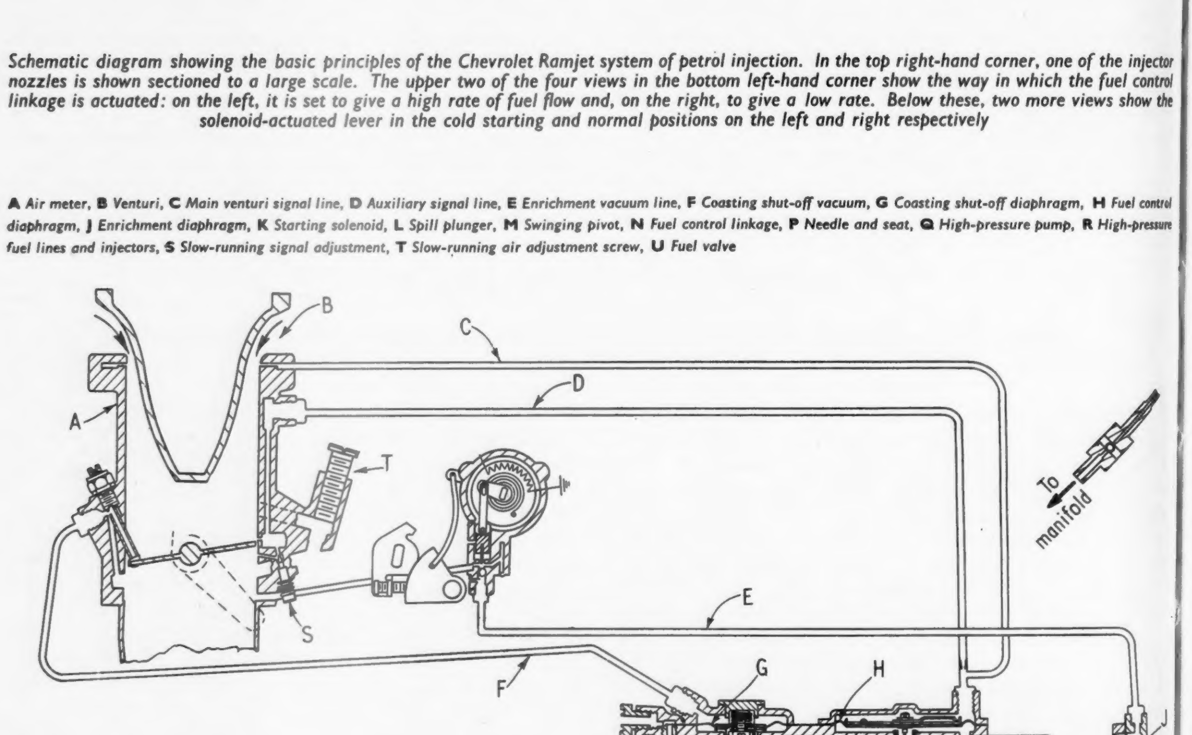
The fuel is injected into this chamber, whence it passes out through a small opening to the induction port. This arrangement has the additional advantage of ensuring that a consistently accurate fuel:air ratio is maintained under idling conditions. The volume of air passing through the chamber, although small when compared with the volume flowing through the intake manifold during normal operation, constitutes a major share of the air used by the engine during idling. At the 0·011 in diameter orifices in the nozzles, the pressure of the fuel varies up to 200 lb/in².

Acceleration

The position of the swinging pivot of the fuel-metering lever is regulated by a rod connected to a fuel enrichment diaphragm. This pivot is normally held in a position that gives maximum economy of operation. The enrichment diaphragm is controlled by the vacuum caused by the flow of air past the throttle valve. At all except large throttle openings, the vacuum in the tube is strong enough to hold the diaphragm back against the opposing force of a spring. This holds the movable pivot in the fuel economy position.

Schematic diagram showing the basic principles of the Chevrolet Ramjet system of petrol injection. In the top right-hand corner, one of the injector nozzles is shown sectioned to a large scale. The upper two of the four views in the bottom left-hand corner show the way in which the fuel control linkage is actuated: on the left, it is set to give a high rate of fuel flow and, on the right, to give a low rate. Below these, two more views show the solenoid-actuated lever in the cold starting and normal positions on the left and right respectively

A Air meter, B Venturi, C Main venturi signal line, D Auxiliary signal line, E Enrichment vacuum line, F Coasting shut-off vacuum, G Coasting shut-off diaphragm, H Fuel control diaphragm, J Enrichment diaphragm, K Starting solenoid, L Spill plunger, M Swinging pivot, N Fuel control linkage, P Needle and seat, Q High-pressure pump, R High-pressure fuel lines and injectors, S Slow-running signal adjustment, T Slow-running air adjustment screw, U Fuel valve



As the throttle control is opened, the vacuum in the connection to the enrichment diaphragm is reduced. The spring then moves the diaphragm in the opposite direction. As a result, the movable pivot is pushed towards the end of the lever, moving the plunger down. Therefore, the rate of fuel return to the reservoir is reduced and the fuel flow to the injection nozzles increased, giving a richer mixture.

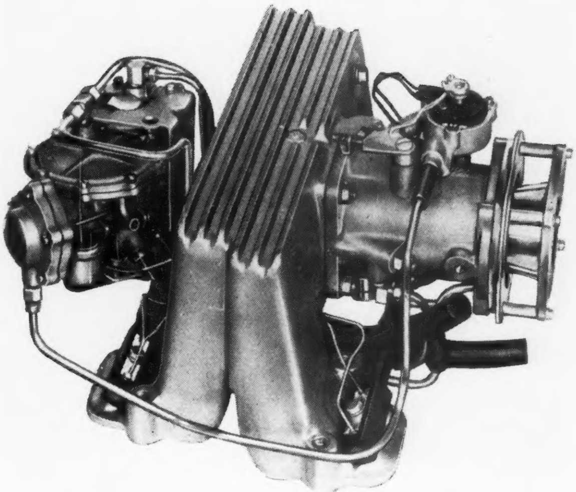
Cold starting

When the starter motor is switched on, it is necessary to get fuel to the nozzles quickly since, at cranking speed, it would take from 20 to 30 sec for the fuel pump to build up enough pressure to unseat the ball check valve. Therefore, a solenoid is used to open a direct fuel passage connected to the nozzles.

When the starter motor is switched on, the solenoid, which is simultaneously energized, forces a connecting link upwards. This, in turn, actuates the starting lever in the metering unit, forcing the plunger down and thus unseating the ball check valve. Fuel then flows directly from the fuel pump to the nozzles. When the driver releases the key type starter switch, the solenoid is de-energized.

After starting, and during warm-up, it is desirable to furnish slightly richer mixtures than would be supplied under normal running conditions. This is accomplished by changing the position of the pivot in the fuel-metering system. A fuel enrichment device and an electric automatic control are used for this purpose. This device is brought into operation by vacuum, from the air meter, acting on the fuel enrichment diaphragm and is regulated by the electric control. From the illustration, it can be seen that the vacuum line, from the air meter to the enrichment diaphragm, is taken through the electric control housing. When the engine is started from cold, vacuum in the housing pulls a check ball upwards against a seat, cutting off the vacuum to the enrichment diaphragm. As a result, the spring actuates the diaphragm and swings the movable pivot towards the end of its lever, moving the plunger down and thus increasing the quantity of fuel supplied to the nozzles.

In the housing, the vacuum is also applied to the lower end of a small piston. The upper end of this piston is linked to a thermostat, which is heated by an element that carries electric current whenever the ignition switch is on. As the thermostat is heated, it relaxes and allows the vacuum to pull the piston downwards. In its lowest position, the piston pushes the check ball off its seat, and thus returns the fuel enrichment system to normal operation. The electric control also actuates a linkage connected to a cam to hold the throttle valve slightly open for fast engine idling after cold starts. As the thermostat heats up, the linkage rotates the cam so that the engine idle speed returns to normal.



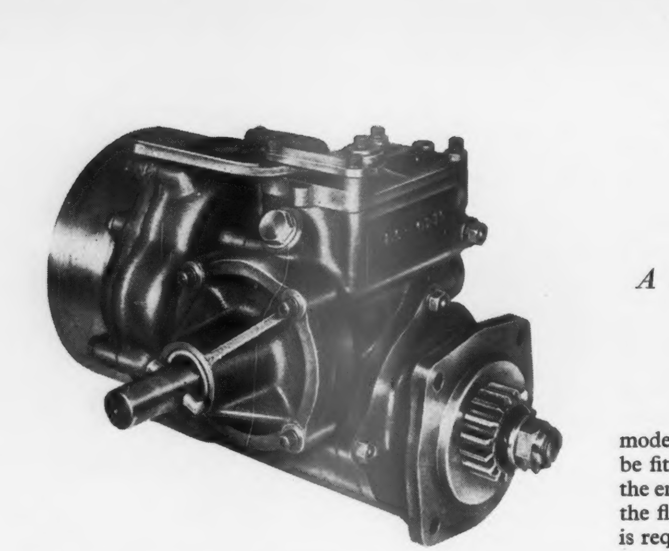
In this illustration, the air meter can be seen on the right and the fuel meter on the left of the induction manifold casting

Coasting

When the vehicle is coasting downhill or decelerating, an automatic fuel cut-off system stops fuel waste and the discharge of exhaust fumes containing unburned fuel. Another advantage obtained by the incorporation of this device is quiet operation of the engine under these conditions. The very high vacuum, obtained when the vehicle is coasting downhill, with the driver's foot off the accelerator, is used to send a signal through a tube to a diaphragm above the high pressure fuel pump. This vacuum raises the diaphragm, and a link connected to it opens a valve in the top of the fuel pump housing, to allow the fuel to be discharged from the pressure side of the pump directly back to the reservoir. As a result, none of the fuel from the pump is supplied to the injection nozzles. The vacuum diminishes as the vehicle slows down, or the throttle is opened; this closes the valve over the fuel pump, allowing the fuel to flow again to the nozzles. Transition from fuel cut-off to normal operation is said to be so smooth that the driver is not aware of it.

The Simms Inertia Starter

A Flange-Mounted, Manually Energized Unit Driving through the Flywheel Ring Gear



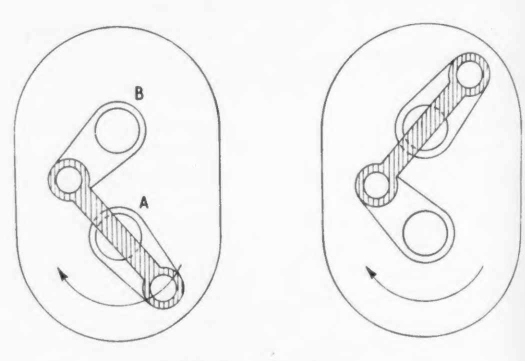
The Simms inertia starter is flange-mounted and can replace a conventional electric starter motor without need for engine modification

ALTHOUGH the conventional electric starter motor for internal combustion engines has been developed to a high state of efficiency and reliability and is in general use, there are numerous applications and operating conditions where electrical installations are unnecessary or undesirable. Examples occur on power units used solely in emergency, at infrequent intervals, or after protracted periods of inactivity. In such instances difficulty may be experienced in maintaining an adequately charged battery.

Climate also exercises a potent influence on starting. Under tropical conditions battery maintenance can become a major problem, and servicing facilities are frequently remote. In extremely cold areas, high starting torques are called for to free the engine initially from the effect of congealed lubricating oil. This duty constitutes a severe drain on the battery and may lead to subsequent inability to effect a start. With either extreme of atmospheric temperatures a manually energized starter can circumvent or obviate the difficulties. In undeveloped areas where labour standards and operator training may be expected to be low, a relatively simple, mechanical starter may present fewer hazards.

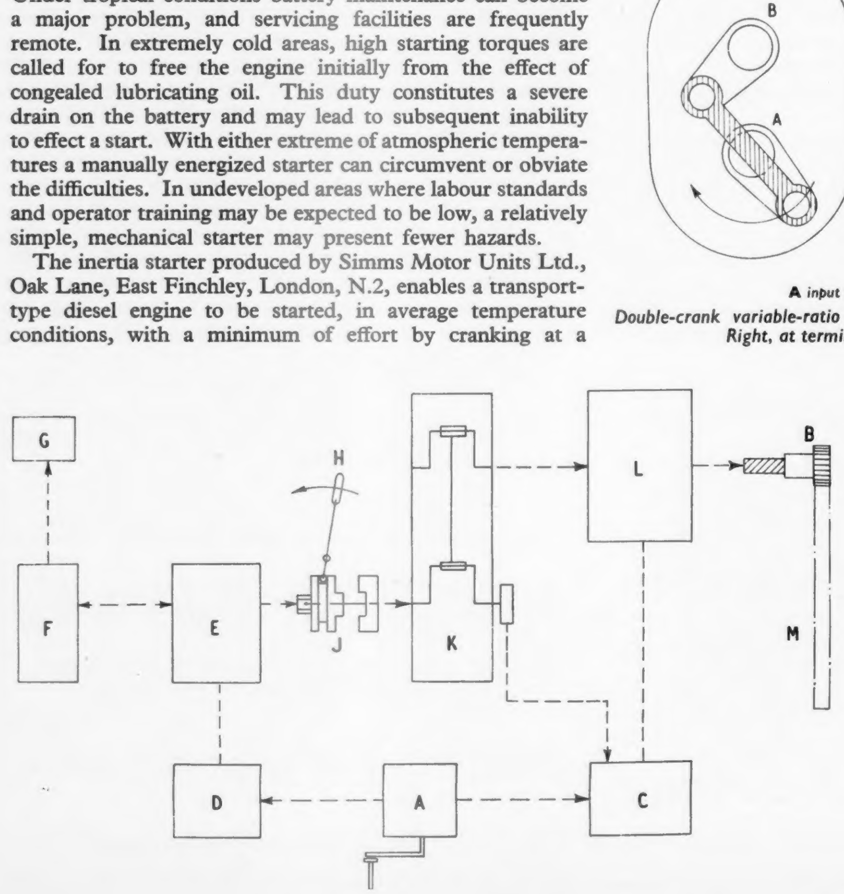
The inertia starter produced by Simms Motor Units Ltd., Oak Lane, East Finchley, London, N.2, enables a transport-type diesel engine to be started, in average temperature conditions, with a minimum of effort by cranking at a moderate speed for not more than ten seconds. It can easily be fitted to replace a standard electric starter, and it rotates the engine by means of a conventional starter pinion engaging the flywheel ring gear in the usual manner. No more space is required and no higher load is imposed on the crankshaft. Thus no modification of the engine is necessary.

A feature of the starter is the unusual variable-ratio transmission device incorporated in the drive from the rotating inertia mass to the starter pinion. This contributes largely to the high efficiency of the unit as it enables the kinetic energy stored in the inertia mass to rotate the engine smoothly from rest without resort to a friction clutch that would dissipate a substantial proportion of the available energy. The device consists of a pair of cranks on parallel spindles; angularly disposed at 90 deg to each other and connected by a link member. When the engagement lever is



A *input crank,* **B** *output crank*

Double-crank variable-ratio transmission. Left, at take-up position. Right, at termination of energy transfer



***A** hand drive, **B** starter pinion, **C** auxiliary drive gear, **D** main drive gear, **E** input reducing gear and winding gear (epicyclic), **F** inertia mass, **G** signal bell, **H** engagement lever, **J** dog clutch **K** variable-ratio transmission, **L** output increasing gear and pinion-engaging gear (epicyclic), **M** engine flywheel ring gear*

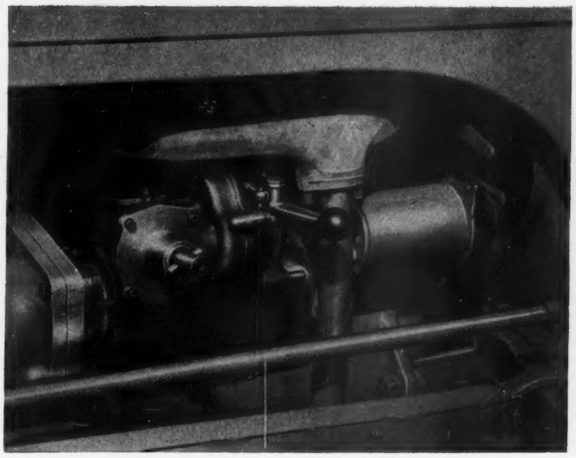
Diagrammatic arrangement of inertia starter

operated the inertia mass, rotating at approximately 8,000 r.p.m., drives input shaft A through reduction gearing. Output shaft B is smoothly accelerated and drives the starter pinion through increasing gearing. Both cranks turn through 180 deg to transfer the stored energy to the engine, and the output crank is smoothly arrested in the position shown in the right-hand diagram. The cranks are reset to the take-up position automatically when cranking is commenced.

The internal arrangement of the gearing in the starter is shown diagrammatically. Immediately the cranking handle is rotated the starter pinion is moved into engagement with the ring gear on the engine flywheel and, during the whole of the cranking period, the engine is slowly rotated through a train of auxiliary gears. This operation helps to free the moving parts of the engine in preparation for a start and also primes the fuel injection system.

After cranking for about ten seconds, the inertia mass is brought up to the appropriate speed of rotation and a bell automatically gives audible warning. This is the signal to cease cranking and to engage, by means of a small hand lever, the dog clutch which couples the inertia mass through its reducing gear to the variable-ratio transmission unit and thence through the increasing gear to the starter pinion. The engine crankshaft is accelerated rapidly for two revolutions and, when ignition occurs and the engine starts, the pinion is disengaged automatically from the ring gear in the usual manner.

Two versions of the starter are produced; types SXA and SXB having the driving pinion on the right and on the left respectively when viewed from the cranking position. The weight of the unit is 73 lb (33 kg) and it is suitable for starting diesel engines having from one to six cylinders of a swept capacity up to approximately 61 in³ (1,000 cc) per cylinder. Inertia flywheel speed is about 8,000 rev/min.



Starter installed on the engine of an agricultural tractor

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

FEBRUARY

Birmingham

Tuesday, 26th February, 6.30 p.m., in the James Watt Memorial Institute, Great Charles Street, Birmingham. Address by the Chairman of the Automobile Division, A. G. Booth, M.B.E. (Member), entitled "Experiences During Forty Years of Automobile Design."

Derby

Monday, 18th February, 7.0 p.m., in The Midland Hotel, Derby. Papers: "The Turbo-Charger on High-speed Diesel Engines—Present Position and Future Prospects," by C. H. Bradbury (Member), and "An Approach to the Problem of Pressure Charging the Compression Ignition Engine," by D. W. Tryhorn, B.Sc. (Associate Member).

North-Eastern

Wednesday, 20th February, 7.30 p.m., in the Chemistry Lecture Theatre, The University, Leeds. Paper: "A Review of Hydrokinetic Fluid Drives and their Possibilities for the British Motor Industry," by J. G. Giles (Associate Member).

North-Western

Thursday, 14th February, 7.15 p.m., in the Engineers' Club, Manchester. Paper: "Brake Usage under Service Conditions," by N. Carpenter (Associate Member).

Scottish

Monday, 18th February, 7.30 p.m., in the Institute of Engineers and Shipbuilders, 39 Elmbank Crescent, Glasgow, C.2. Paper: "Fatigue of Metals," by J. A. Pope, D.Sc., Ph.D., B.Sc. (Member).

Western

Thursday, 28th February, 6.45 p.m., in the Royal Hotel, Bristol. Paper: "Transmission Development for Public Service and Heavy Goods Vehicles," by A. Gordon Wilson (Associate Member).

MARCH

London

Tuesday, 12th March, 6.0 p.m., at 1 Birdcage Walk, Westminster, S.W.1. Automobile Division General Meeting. Paper: "Transmission Development for Public Service and Heavy Goods Vehicles," by A. Gordon Wilson (Associate Member).

Coventry

Tuesday, 5th March, 7.15 p.m., in the Grosvenor Room, Leofric Hotel, Coventry. Series of short papers on engine detail components: "Sealing Rings for Wet Cylinder Liners," by J. L. Hepworth (Member), "Recent Developments and Modern Techniques in the Use of Gaskets," by M. G. Herrington, "The Design and Application of Thermostats to the Internal Combustion Engine Cooling System," by S. H. Blazey (Member), "Gudgeon Pin Location," by R. C. Cross (Member).

Luton

Monday, 11th March, 7.30 p.m., in the Assembly Room, Luton Town Hall. Paper: "A Review of Hydrokinetic Fluid Drives and their Possibilities for the British Motor Industry," by J. G. Giles (Associate Member).

Birmingham

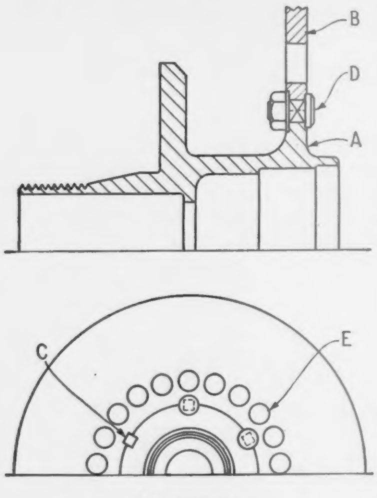
Tuesday, 5th March, 7.0 p.m., in the George Hotel, Stafford Street, Wolverhampton. Paper: "The Free Piston Engine and its Application," by R. W. S. Mitchell, M.Sc. (Member).

CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Detachable brake disc

This method of attaching a brake disc to a wheel hub saves weight and space in comparison with the usual registered and bolted flange assembly. The hub is furnished with a radial flange A, on the periphery of which the brake disc B is mounted with a sliding fit. A plurality of rectangular slots are formed in the periphery of the flange and are radially aligned with complementary slots in the disc. In association, the slots in the assembled



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components form square apertures C to receive the square shanks of the fixing bolts D. The disc is thus axially secured and radially keyed to the flange.

Symmetrically pitched holes E are formed in the disc between the inner periphery and the working surfaces to save weight and to reduce the conduction of heat from the disc to the hub. *Patent No.* 752101. *Dunlop Rubber Co. Ltd.*

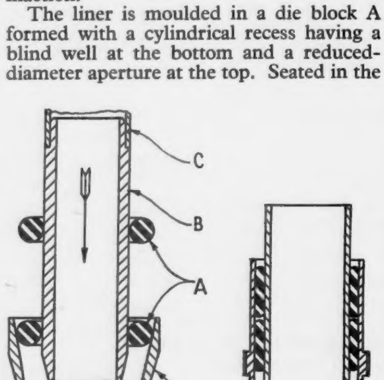
Assembly of resilient bushings

A familiar method of producing resilient bushings is to force rubber rings of rectangular cross section through a die and into the outer metal bush with a sliding motion. With the aid of a conical plug the inner metal bush is then forced into the rubber rings, again with a sliding motion. The contacting surfaces of the metal bushes require an expensive finishing operation and the rubber rings need a lubricant in order to facilitate assembly. To avoid deterioration of the rubber the lubricant should be specially selected and no lubricant should remain between ring and bush, when assembled, or the functioning of the bushing could be impaired. Assembly is a relatively slow process and difficulties are likely to be encountered in the assembly of lengthy parts or parts subjected to relatively high torque values.

These disadvantages are obviated by the method of the invention. The rubber rings are of circular cross-section and may be fitted in a part-vulcanized condition and over-vulcanized after assembly. Contacting surfaces on the metal bushes are required to be rough machined only.

The two rubber rings A are placed on a cylindrical plug B, one end of which is of reduced diameter to receive the inner metal bush C. A hollow die D is formed as a truncated cone and is recessed to accommodate the end of the outer metal bush E. By exerting pressure in the axial direction indicated by the arrow, the rings A are displaced by a rolling action and bush C is entered into bush E. In the process the rings are progressively deformed to assume the flattened form shown in the assembled unit. Fitting is effected without the use of lubricant and the coefficient of friction between rubber and metal is fully maintained.

An apparent disadvantage in the use of circular-section rubber rings is the reduction of resilient resistance to axial forces. This is turned to an advantage by the expedient of fitting the rings in a partly vulcanized condition. When finally vulcanized the rubber fibres are stabilized in the deformed state and the resulting axial resistance is greater than is obtained by conventionally assembled square-section rings. *Patent No.* 752230. *S. A. Andre Citroën (France).*

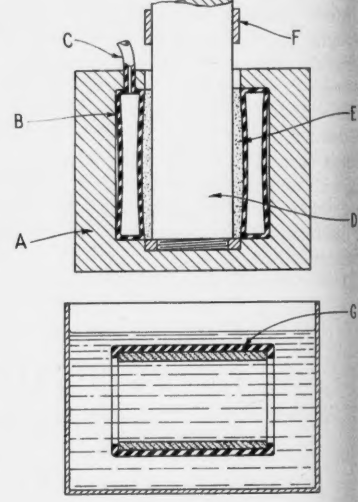


No. 752230

Lubricant-retaining cylinder liner

This cylinder liner may be of iron or a phosphor bronze and is produced by powder metallurgy techniques. The outer surface and the major portion of the wall section is imperforate but the inner surface is porous to a predetermined depth. The size of the pores are controlled by particle size and briquetting pressure so as to obtain a capillary action which will draw in lubricant. Such a lubricated bore surface, it is claimed, will resist wear, prevent ring seizure, and have a longer useful working life than a solid metal surface. The lubricant will be retained in the pores even after prolonged periods of inaction.

The liner is moulded in a die block A formed with a cylindrical recess having a blind well at the bottom and a reduced-diameter aperture at the top. Seated in the recess is a flexible moulding bag B having a hollow stem C through which a pressure fluid can be introduced to inflate the bag. A centre post D with a detachable end flange which seats in the well in block A is, thereby, centred in the recess. Powdered metal E is poured through the aperture while the bag is deflated and the sleeve punch F is then lowered to tamp the powder and to close and seal the mould cavity. Liquid at a pressure of approximately 3,000 lb/in^2 is then introduced into



No. 751649

the bag B to compact the powder into a coherent briquette in the shape of the cylinder liner.

After deflating the bag, the compact is stripped from the mould by withdrawing centre post D and, after detaching the end flange, it is stripped from the post by punch F. The compact is then placed vertically, with a ring of copper on the upper end, in a furnace heated to a temperature of from 1,125 to 1,150 deg C for a period of from 1 to 2 hours. This period is sufficient to sinter the compact and allow the melted copper to infiltrate all the voids. When cooled the liner is non-porous throughout.

The liner is then inserted in a rubber sleeve G, covering all but the bore surface, and placed in a tank containing an acid solution that will selectively dissolve out the copper from the iron. It is held in the solution for a period sufficient to dissolve out the copper to a depth between 0·001 and 0·004 in to leave the desired capillary passages between the metal particles. *Patent No.* 751649. *Thompson Products Inc. (U.S.A.).*

CORRECTION

It is regretted that in the review of Patent No. 750252 relating to a rubber glazing strip, which appeared in the January issue, the owners of the Patent were incorrectly described. The name of the Company should be "Windshields of Worcester, Ltd."

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